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Spiral tube heat exchanger for liquid metal

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SPIRAL TUBE HEAT EXCHANGER

FOR LIQUID METAL

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W. Yatchmenoff

SPIRAL TUBE HEAT EXCHANGER FOR LIQUID METAL

by

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[redacted]

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Approved: [redacted]

Academic Dean (acting)

PREFACE

The developement of a heat transfer conversion system utilizing liquid metals has been the object of the investigations with which I have been associated on my present duty with the General Electric Company, Schenectady, New York. The major problems involved have been the design of the components of the system. One of the major components under developement is a heat exchanger for liquid metals.

The developement of a suitable heat exchanger has been accomplished by contracting with various companies for the independent design and in some cases construction for testing of various types of exchangers. The spiral tube type exchanger which I have chosen to investigate, though probably not original in design, was not included in the types chosen for investigation. As I believe that this type of exchanger shows promise as being suitable for this application, I have chosen to investigate its possibilities. The investigation of this exchanger is the subject of this paper.

I should like to acknowledge the assistance and guidance of Captain H. Burris USN who is in charge of the power conversion system project on which I have been working. The various Engineers of the General Electric Company with whom I have been associated on this project have been of assistance in advising on various technical aspects of this investigation. Assistance on the heat transfer problems was provided by Associate Professor Harold A. Johnson of the University of California on loan to the General Electric Company.

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TABLE OF SYMBOLS AND ABBREVIATIONS

ρ	Density
μ	Viscosity
a	Thermal Diffusivity
c	Specific Heat
f	Fanning Friction Factor
k	Thermal Conductivity
h	Fluid Coefficient of Heat Transfer
u	Wall Thickness
C	Tube Circumference
D	Hydraulic Diameter
F	Force
G	Weight Rate per Unit Area
M	Bending Moment
P	Pressure
Q	Thermal Capacity, Heat Rate
S	Stress
V	Volume Rate of Flow
W	Weight Rate of Flow
Nu	Nusselt's Number
Re	Reynold's Number
Pr	Prandtl's Number

INTRODUCTION

In the never ending search for means by which the power-weight or power-size ratio for power plants may be increased, liquid metals are assuming increasing importance. Their excellent heat transfer characteristics make them most suitable as heat transfer media to accomplish either of two purposes oft times essential to a given power plant. More efficient means of cooling allows higher working temperatures to be utilized and thusly, efficiencies increased and size and weight in many cases to be reduced. The use of sodium cooled exhaust valves in internal combustion engines is cited as one of the first and simplest applications of liquid metals to this purpose.

The second use of liquid metals as heat transfer media is in those applications which require the transfer of heat from one location to another not for purposes of cooling but as a carrier of energy. The properties of most liquid metals which suit them to this purpose are; high coefficient of heat transfer, reasonably high heat capacity, and relatively low viscosity.

In this case of heat transfer from one location to another, the problems involved are mainly the design of the various components of the system. The design of one of these components is the project I have chosen to

illustrate in this paper. Namely, the design of a heat exchanger involving liquid metals as fluids. The type of exchanger chosen for investigation is of the spiral tube type illustrated by Figures 21, 22, 23, and 24 on pages 61, 62, 63, and 64 using counter flow of the two fluid streams. One fluid enters at the center and spirals outward and the other enters at the outer end and spirals inward.

The results of the investigation show that this type exchanger is suitable for the application. All specifications as set up in Chapter I are met with an exchanger using 270 tubes for each fluid. The tubes are rectangular, 1/4 inch by 5/8 inch with an 0.020 inch tube wall. The total volume occupied by the exchanger, including headers, is 25 cu. ft., which compares very favorably with more conventionally designed heat exchangers which occupy on the average, a volume of about 30 cu. ft.

CHAPTER I

THE PROBLEM

1. Objective

The design of the liquid metal heat exchanger, which has been chosen for illustration in this paper, has as its objective the determination of the minimum volume of a spiral tube type heat exchanger meeting the specifications which follow.

2. Specifications

a. Fluids are liquid sodium

b. Exchanger to be constructed of type 18-8 stainless steel

c. Maximum pressure drop for each fluid limited to 15 psi.

d. Capacity of exchanger is 100×10^6 BTU/hr

e. Hot liquid inlet temperature is 700°F and outlet is 400°F

f. Cold liquid inlet temperature is 350°F and outlet is 650°F

g. Tube pressure will not exceed 50 psi

h. Shell designed to withstand 300 psi

i, The space within the shell and between the tubes will be filled with sodium

3. Method of attack

A certain amount of preliminary work must be done

before any attempt is made to determine the the final design. This preliminary work will include the determination of certain physical properties of sodium and 18-8 stainless steel. A type of central header design must be selected and a method of fabricating the tube to header joint must be determined.

After the above preliminaries, a determination of the optimum tube size and shape will be made. This will be accomplished in the following manner:

a. A size of square tube will be selected.

b. The volume required for an exchanger using a given number of tubes will be determined.

c. Step b will be repeated for other numbers of tubes.

d. A plot of the results of b and c will give the minimum volume to be obtained for this size square tube.

e. Steps b, c, and d are to be repeated for other tube sizes.

f. A plot of the minimum volumes obtained in step e will give the minimum volume of exchanger to be obtained using square tubes.

g. Using the same area of tube cross section as the optimum square tube in step f, calculations will be made for rectangular tubes. Variations in tube area may be made if necessary to determine the most desirable tube.

The above calculations will be made utilizing various simplifying assumptions which will allow simpler means of determining the correct tube size and shape

though not exactly the correct dimensions for the final design. Therefore, after determining the optimum tube size and shape and number of tubes, more accurate calculations will be necessary to determine the final design.

4. Physical Properties

As a result of a search through available literature, the following values of various physical constants of sodium and 18-8 sta inless steel have been chosen for use:

Properties of sodium

a. Density, Rinck, E. (9)

Temperature (^{o}E)	Density, ρ (lbs/cu ft)
300	57.2
400	56.3
500	55.5
600	54.6
700	53.8

b. Viscosity, Handbook of Chemistry and Physics (5)

Temperature (^{o}F)	Viscosity, μ (lbs/hr ft)
300	1.32
400	1.07
500	0.91
600	0.80
700	0.71

c. Specific Heat, Handbook of Chemistry and Physics (5)

Temperature (^oF)	Specific Heat, c (BTU/lb ^oF)
300	0.331
400	0.335
500	0.339
600	0.342
700	0.344

d. Thermal Conductivity, Hall, W.C. (4)

Temperature (^oF)	Thermal Cond., k (BTU/ft^2 hr ^oF/ft)
300	47.2
400	45.5
500	44.0
600	43.1
700	42.1

For the purpose of simplifying the initial calculations, the following constant values for the preceding physical properties will be used. The accuracy of the results will be within a few percent of the correct values. Corrections will be made in the final calculations to give more accurate results.

Density, (lb/cu ft)	55.2
Viscosity, (lb/hr ft)	0.86
Specific Heat, (BTU/lb ^oF)	0.340
Thermal Conductivity, (BTU/ft^2 hr ^oF/ft)	43.7

Properties of 18-8 stainless steel, type 347 (17/19 Cr, 8/12 Ni, 2.00 max. Mn, 0.10 max. C), Hoyt, S.L. (6), American Society for Metals (1), Uhlig, C. (10).

a. Tensile Strength 80,000-90,000 psi

b. Yield Strength 35,000-40,000 psi

c. Modulus of Elasticity 28.5×10^6 psi

d. Coefficient of Expansion $10.2 \times 10^{-6}/{}^{\circ}F$ in range of $32^{\circ}F$ to $932^{\circ}F$

e. Thermal Conductivity 11 $BTU/ft^2\ hr\ {}^{\circ}F/ft$

CHAPTER II

GENERAL DESIGN OF CENTRAL HEADERS

1. Header Type

Figure 1, page 9, is a sketch of the type of central header to be used. The area of the larger end of each header should be approximately the same as that of the service piping which in this case is ten inch pipe. With about a five inch minimum width on the small end required to allow internal access, the overall diameter of the headers must be approximately fourteen inches which gives an area of header at the large end of about 80 square inches as compared to about 78 square inches for the pipe.

2. Welding and Brazing of Stainless Steel

For the problem in question, the techniques of welding and brazing type 347 stainless steel need not be investigated further than determining methods which will be suitable for our purposes.

Manganese-nickel brazing alloy appeared to be a likely alloy and was chosen for investigation. Tests resulted in the selection of the eutectic mixture of 60% manganese-40% nickel as being the most suitable composition, and acceptable for this application. The use of a suitable flux is required to insure proper wetting of the surfaces. The brazing temperature is 2100 to 2200 $^{\circ}$F. The thin walls of the tubing make the use of a salt bath advisable in heating to the brazing temperature.

CENTRAL HEADER TYPE

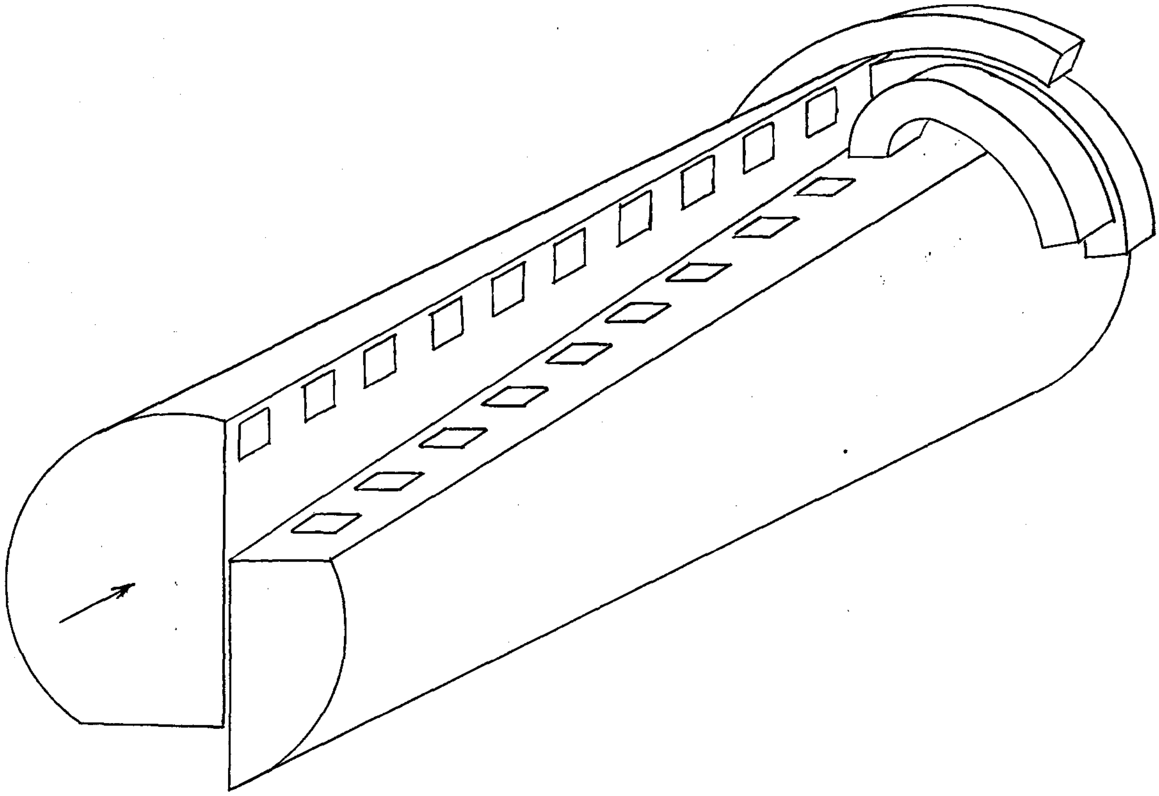


Figure 1

Welding of stainless steels has become commonplace in industries using these materials and is therefore of no particular problem. Satisfactory welds are obtained by changing the normal direction of potential and using a gas shield during the welding operation with a stainless rod of composition similiar to the material to be welded.

3. Tube and Header Construction

Figure 2 is an illustration of the type of tube to header joint to be used. The sleeves are brazed to the ends of the tubes prior to installation. The sleeves are welded to the headers with the weld being made on the inside of the header. All tubes are joined to the central headers before the outer headers are set in place and the tube sleeves welded thereto.

Assembly of the tubes to the central headers is started at one end with each of the two tubes of each consecutive row being set in place and welded. The curved portions of the headers are made of about one foot sections which are welded in place as the preceding sections are filled with tubes. This procedure allows the welder to work no further than one foot from the end of the section.

HEADER JOINT

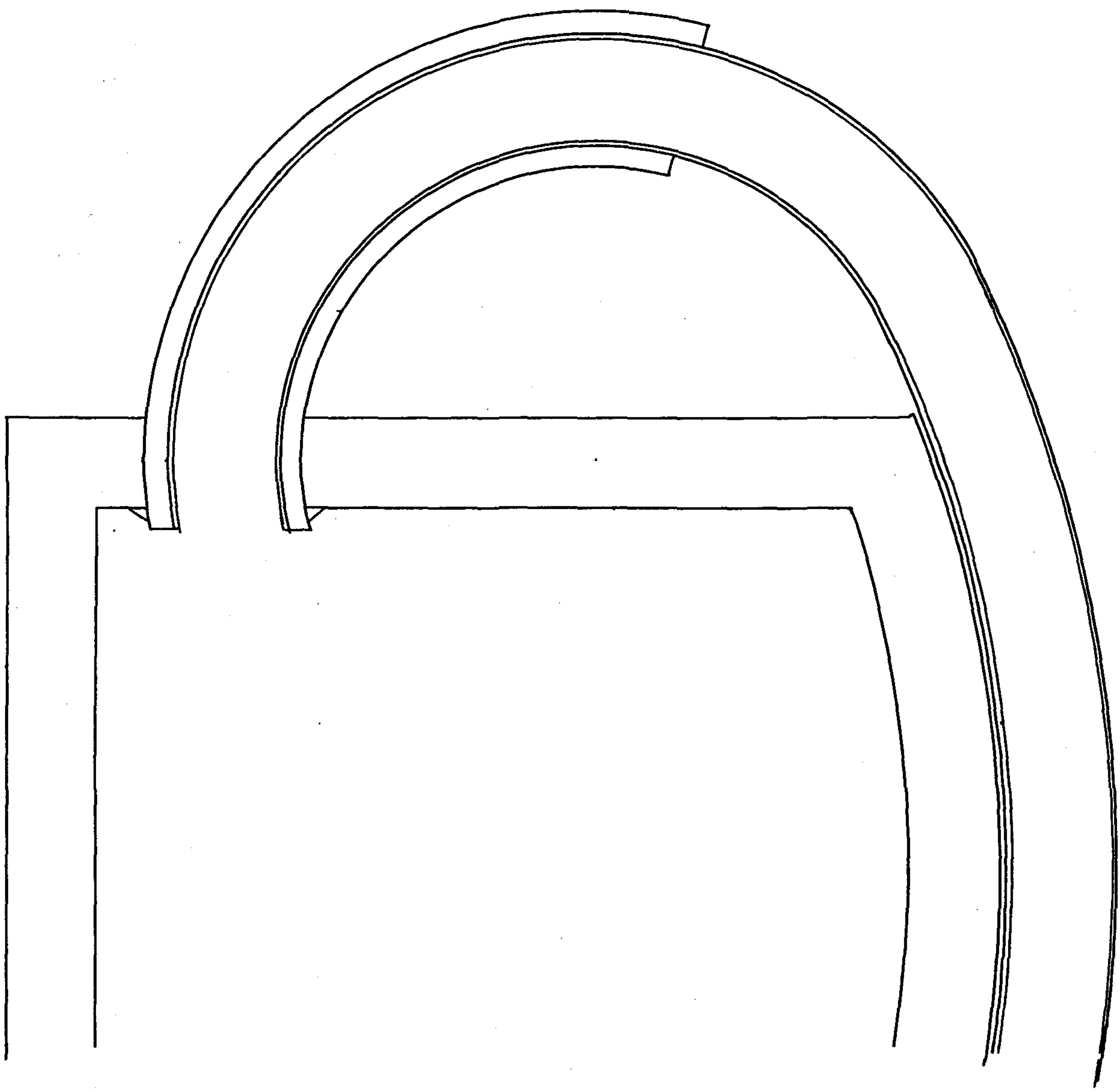


Figure 2

CHAPTER III

TUBE SIZE

1. Selection of First Tube Size

The total volume of the exchanger is estimated to occupy about 40 cubic feet. A reasonable shape would be with the height equal to the diameter. This results in a diameter of about 45 inches.

With the above estimate of size and assuming a 2 psi pressure drop to occur at the tube inlet and outlet and allowing one velocity head for tube inlet and tube outlet, we can arrive at a very rough estimate of the tube size.

$$Q = W_h c(T_{h1} - T_{h2})$$

Q capacity (BTU/hr)

W_h rate of flow, hot fluid (lb/hr)

c specific heat (BTU/lb ^oF)

T_{h1} inlet temperature, hot fluid (^oF)

T_{h2} outlet temperature, hot fluid (^oF)

$$100 \times 10^6 = W_h 0.340(700-400)$$

$$W_h \doteq 0.98 \times 10^6 \text{ (lb/hr)}$$

$$V_h = \frac{W_h}{\rho}$$

V_h volume rate of flow, hot fluid (cu ft/hr)

ρ density (lb/cu ft)

$$V_h = \frac{0.98 \times 10^6}{55.2}$$

$$V_h = 17{,}700 \text{ (cu ft/hr)}$$

With the assumption of a 2 psi pressure drop corresponding to two velocity heads for entrance and exit loss, we have the following;

$$\frac{1}{144}\,\frac{2v^2\rho}{2g} = 2$$

$$v = 12.9 \text{ ft/sec}$$

$$V_h = vA$$

$$17{,}700 = 12.9 \text{x} 3600\ A$$

$$A = 0.327 \text{ ft}^2,\ 47.2 \text{ in}^2$$

For a height of 45 inches, the size of tube would be;

$$\frac{47.2}{45} = 1.05 \text{ inch square tube}$$

Therefore, for the first tube size, a one inch inside dimension square tube will be used.

2. Tube Wall Thickness

A design stress of 30,000 psi will be used allowing a factor of safety of approximately 20% for a yield strength of 35,000 to 40,000 psi.

Maximum bending moment (at corner)

$$M = \frac{wl^2}{12}$$

$$M = \frac{50 \text{x} 1^2}{12}$$

$$M = 4.16 \text{ lb in}$$

Maximum tension force (at corner)

$$F = PA$$

$$F = \frac{50 \times 1 \times \sqrt{2}}{2}$$

$$F = 35.4 \text{ lb}$$

Max. Stress = Max. Tensile Stress + Max. Bending Stress

$$S = S_T + S_M$$

Tension

$$S_T = \frac{F}{u} = \frac{35.4}{u}$$

u wall thickness

Bending

$$S_M = \frac{Mc}{I} = \frac{M x u/2}{u^3/12} = \frac{6M}{u^2}$$

$$S_M = \frac{6 \times 4.16}{u^2}$$

$$30{,}000 = \frac{35.4}{u} + \frac{6 \times 4.16}{u^2}$$

u = 0.0293 inches wall thickness

CHAPTER IV

HEAT TRANSFER

1. Simplifying Assumptions

Figure 3, page 16, is a sketch which approximates the exchanger when considering the center tube in the sketch. This sketch illustrates the appearance relative to the center tube should the tube spirals be unrolled and will assist in explaining the assumptions made herein to simplify the calculations. It is recognized that although the results may be in error by several percent we are able to determine the size and shape of tube giving the minimum overall volume of the exchanger. When the selection of the tube for use has been made, the final calculations will be made in a more accurate manner.

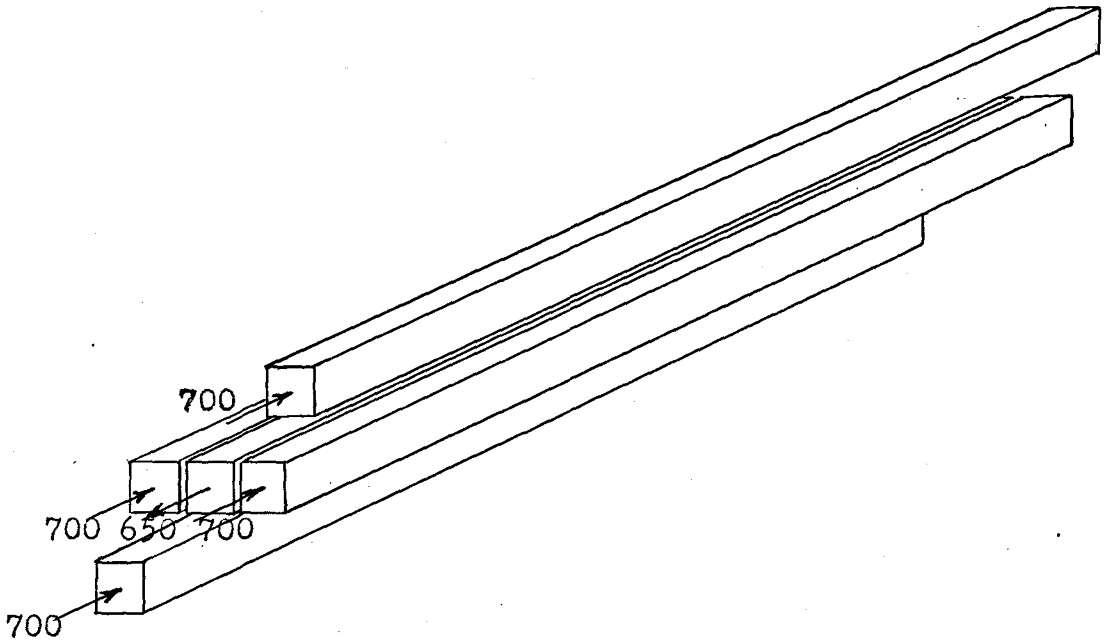
The temperature change along the tube is assumed to be linear. With this assumption, we arrive at the conclusion that at any point along the tube, the temperature differences between tubes in the radial direction differ from the temperature differences in the longitudinal direction by the same amount. Referring to figure 3, we have the following;

$$(T_{h+}-T_c) - (T_h-T_c) = (T_h-T_c) - (T_{h-}-T_c)$$

$$(T_{h+}-T_c) + (T_{h-}-T_c) = 2(T_h-T_c)$$

As the major temperature drop is across the tube walls and the flow is turbulent, the temperature across

SCHEMATIC OF SIMPLIFIED TUBE ARRANGEMENT



CROSS SECTION

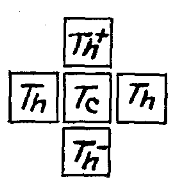


Figure 3

the tube is considered to be constant. In an element of length dl, the rate of heat transfer to the colder tube (T_c) will conform to the following;

$$dq = UdA(\Delta T)$$

$$dq = 2Ubdl(T_h - T_c) + Uadl(T_{h+} - T_c) + Uadl(T_{h-} - T_c)$$

a tube thickness (radial direction)

b tube width (longitudinal direction)

$$dq = 2Ubdl(T_h - T_c) + Uadl\left[(T_{h+} - T_c) + (T_{h-} - T_c)\right]$$

From the previous temperature relationship,

$$dq = 2Ubdl(T_h - T_c) + Uadl\left[2(T_h - T_c)\right]$$

$$dq = Udl(T_h - T_c)2(b+a)$$

$$dq = UCdl(T_h - T_c)$$

C tube circumference

For the total heat transferred to the tube,

$$\int_0^q dq = \int_0^L UC(T_h - T_c)dl$$

$$q = UCL(T_h - T_c)$$

2. Heat Transfer Coefficients

Using the above derived expression, we may determine the length of tube and thusly the overall size of the heat exchanger once we have established the value of the overall heat transfer coefficient, U. The reference area for the evaluation of the coefficients comprising the overall coefficient is taken to be the inner area of the rectangular tube.

The overall resistance being the sum of the resistances of each part of the path of heat flow, the following

expression holds;

$$\frac{1}{U} = \frac{1}{h} + \frac{u}{k_{tube}} + \frac{\text{tube space}}{k_{sodium}} + \frac{u}{k_{tube}} + \frac{1}{h}$$

$$\frac{1}{U} = \frac{2}{h} + \frac{2u}{k_{tube}} + \frac{\text{tube space}}{k_{sodium}}$$

In order that U be as large as possible, the tube spacing must be held to a minimum. The spacing should be no more than 0.010 inches. All other costituents of the equation but the fluid coefficient, h, are known. Therefore, the evaluation of the fluid coefficient is the one remaining problem before we are able to use the heat transfer equations previously derived.

3. Fluid Coefficient of Heat Transfer

The high values of heat transfer coefficients for heat transfer to molten metals is one of the reasons for their ever increasing use. As for all newly used materials, very little experimental data is to be found whereby the commonly used coefficients used in design work may be evaluated. In this case, we turn to the work of R. C. Martinelli (7) wherein the analogy between momentum and heat transfer is used to arrive at an expression whereby the fluid coefficient of heat transfer may be evaluated. This expression is as follows;

$$Nu = 7 + 0.05(Re \cdot Pr)^{0.5} + 0.023(Re \cdot Pr)^{0.8}$$

$Nu = \frac{hD}{k}$ Nusselt's Number

$Re = \frac{vD}{\nu}$ Reynold's Number

$Pr = \frac{\nu}{a}$ Prandtl's Number

D hydraulic diameter

a thermal diffusivity

However, for the number of calculations which must be made herein, to use this expression would be too time consuming in that it would be necessary to evaluate the dimensionless Reynold's and Prandtl's numbers, determine Nusselt's number from the expression of Martinelli, and then determine the heat transfer coefficient, h, from the value of Nusselt's number (hD/k). In order to expedite the work, we make use of the constant values chosen for the physical properties of sodium and from Martinelli's expression, plot hD against GD, figure 4, where G is the weight rate of flow per unit area. Use of this plot, allows the value of h to be easily and quickly determined.

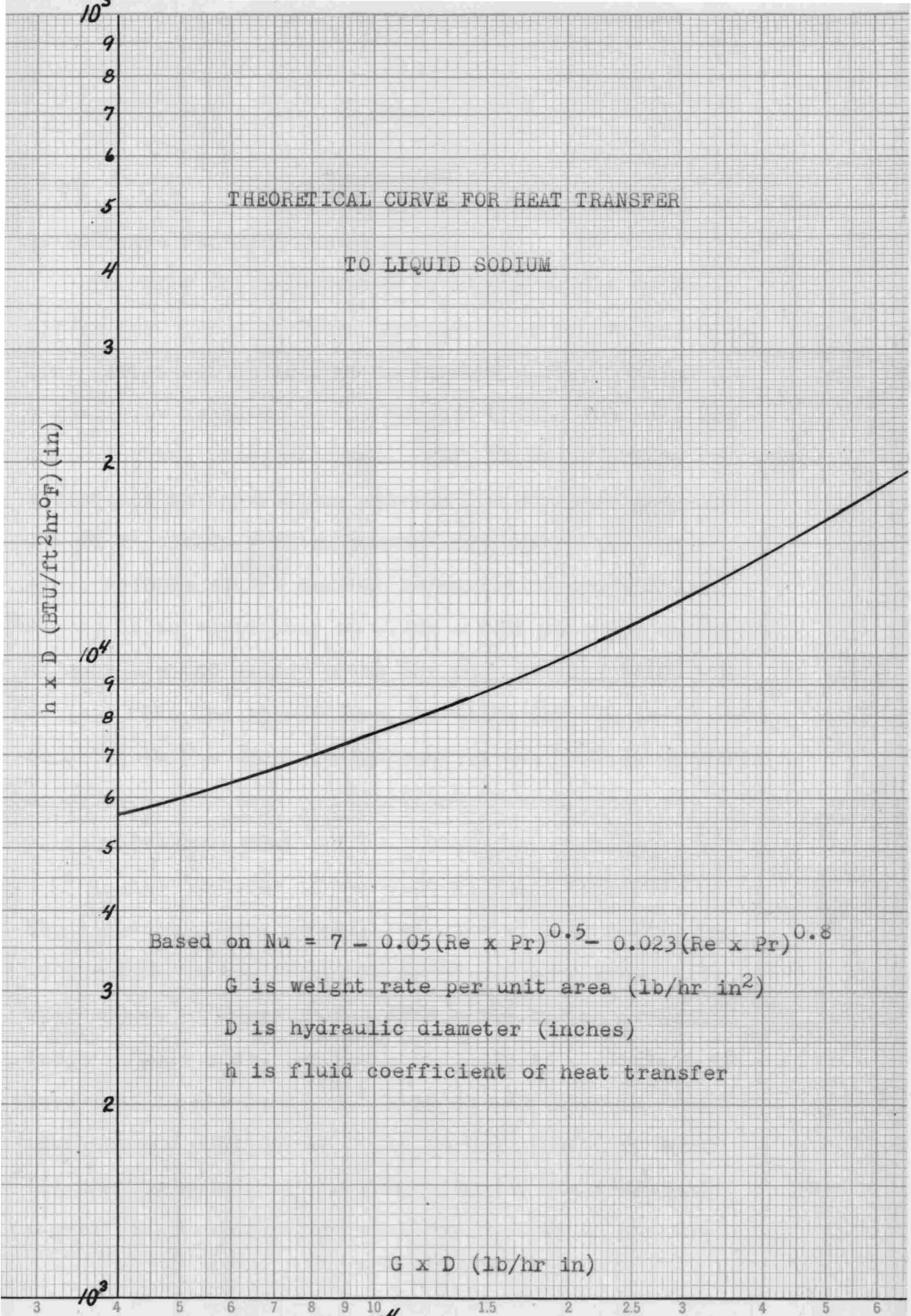

THEORETICAL CURVE FOR HEAT TRANSFER
TO LIQUID SODIUM
h x D (BTU/ft²hr°F)(in)
Based on Nu = 7 – 0.05(Re x Pr)^0.5 – 0.023(Re x Pr)^0.8
G is weight rate per unit area (lb/hr in²)
D is hydraulic diameter (inches)
h is fluid coefficient of heat transfer
G x D (lb/hr in)
10^5
10^4
10^3
10^4

Figure 4

CHAPTER V

FLUID FLOW

1. Entrance and Exit Losses

The flow characteristics of liquid metals do not differ from those of other liquids as is to be expected. This fact is of assistance in testing a final design or a sample by using an easily handled liquid and actually measuring the pressure drops. But, it is of little assistance in determining exactly the entrance and exit losses in design work on unusual configurations. However, working with a maximum allowable pressure drop, we can do no worse than to allow a full velocity head loss at entrance and exit. Using the full losses, which will be less than the true values, we allow for the small loss occuring in the small radius bend in the tube at entrance or exit.

2. Tube Pressure Drop

As the diameter of the tube is small compared to the diameter of the spiral, the tube will be considered as straight in the calculations of pressure drop. The vast amount of experimental data available on pressure drops in pipe flow, permits a much closer evaluation of head loss in the tube than could be done for the tube entrance and exit losses. The curves plotted by Moody (8) are the result of the summation and evaluation of a great many experimental results obtained by various

persons~~s~~ in this field.

The curve to be used in this case if that for very smooth tubing. This curve is approximated by the expression;

$$f = \frac{0.046}{Re^{0.2}}$$

The total pressure drop, P, in passing through the heat exchanger is the sum of the entrance losses, the tube losses, and the exit losses.

$$P = \frac{1}{144}\left(\frac{v^2\rho}{2g} + \frac{fLv^2\rho}{D2g} + \frac{v^2\rho}{2g}\right)$$

$$P = \frac{v^2\rho}{144g}\left(1 + \frac{fL}{2D}\right)$$

CHAPTER VI

SAMPLE CALCULATIONS

1. Heat Transfer Coefficients

For this sample calculation on the one inch square tube, the number of tubes is taken as forty. This gives a tube bundle length of 42.7 inches.

Weight rate per unit area, G,

$$G = \frac{Wh \ (lb/hr)}{A \ (sq\ in)}$$

$$G = \frac{0.98x10^6}{40}$$

$$G = 2.45x10^4 \ (lb/hr\ in^2)$$

Hydraulic diameter, D,

$$D = \frac{4A \ (sq\ in)}{C \ (in)}$$

$$D = \frac{4x1}{4}$$

$$D = 1 \text{ inch}$$

GD

$$GD = 2.45x10^4 \ (lb/hr\ in)$$

hD, from figure 4,

$$hD = 1.09x10^4 \ (BTU/ft^2 hr^oF/in)$$

Fluid coefficient of heat transfer, h,

$$h = \frac{hD}{D}$$

$$h = \frac{1.09x10^4}{1}$$

$$h = 1.09x10^4 \ (BTU/ft^2 hr^oF)$$

Overall coefficient of heat transfer, U,

$$\frac{1}{U} = \frac{2}{h} + \frac{2u}{k_{tube}} + \frac{\text{tube space}}{k_{sodium}}$$

$$\frac{1}{U} = \frac{2}{1.09\text{x}10^4} + \frac{2\text{x}0.0293}{12\text{x}11} + \frac{0.010}{12\text{x}43.7}$$

$$U = 1560 \text{ (BTU/hr ft}^{2\,o}\text{F)}$$

2. Exchanger Volume

Tube length, L,

$$q = UCL(T_h - T_c)$$

$$L = \frac{q}{UC(T_h - T_c)}$$

$$L = \frac{100\text{x}10^6\text{x}12}{1560\text{x}4\text{x}50\text{x}40}$$

$$L = 96 \text{ ft}$$

Tube bundle diameter

The 96 feet of tube would require $10\frac{1}{4}$ revolutions of the spiral. The diameter of the tube bundle would therefore be 57.5 inches.

Tube bundle volume

$$V = \frac{\pi D^2 H}{4}$$

$$V = \frac{\pi(57.5/12)^2\text{x}(42.7/12)}{4}$$

$$V = 64.2 \text{ ft}^3$$

Exchanger volume

The total volume of the exchanger is the volume of

the tube bundle plus the volume of the outer headers. The discharge ends of the headers should have an area equal to that of the ten inch service piping. In this case the outer header volume is 1.9 cu ft. Therefore, the total volume of the exchanger is 64.2 + 1.9 = 66.1 cu ft.

3. Pressure Drop

Velocity, v,

$$v = \frac{Vh}{A}$$

$$v = \frac{17,700x144}{60x60x40}$$

$$v = 17.8 \text{ (ft/sec)}$$

Fanning friction factor, f,

$$f = \frac{0.046}{Re^{0.2}}$$

$$Re = \frac{Dv\rho}{\mu}$$

$$Re = \frac{(1/12)17.8x55.2x3600}{0.86}$$

$$Re = 343,000$$

$$f = \frac{0.046}{343,000^{0.2}}$$

$$f = 0.00359$$

Pressure drop, P,

$$P = \frac{\rho v^2}{144g}(1 + \frac{fL}{2D})$$

$$P = \frac{55.2x17.8^2}{144x32.2}(1 + \frac{0.00359x96}{2x1/12})$$

$$P = 11.6 \text{ psi}$$

CHAPTER VII

SQUARE TUBE RESULTS

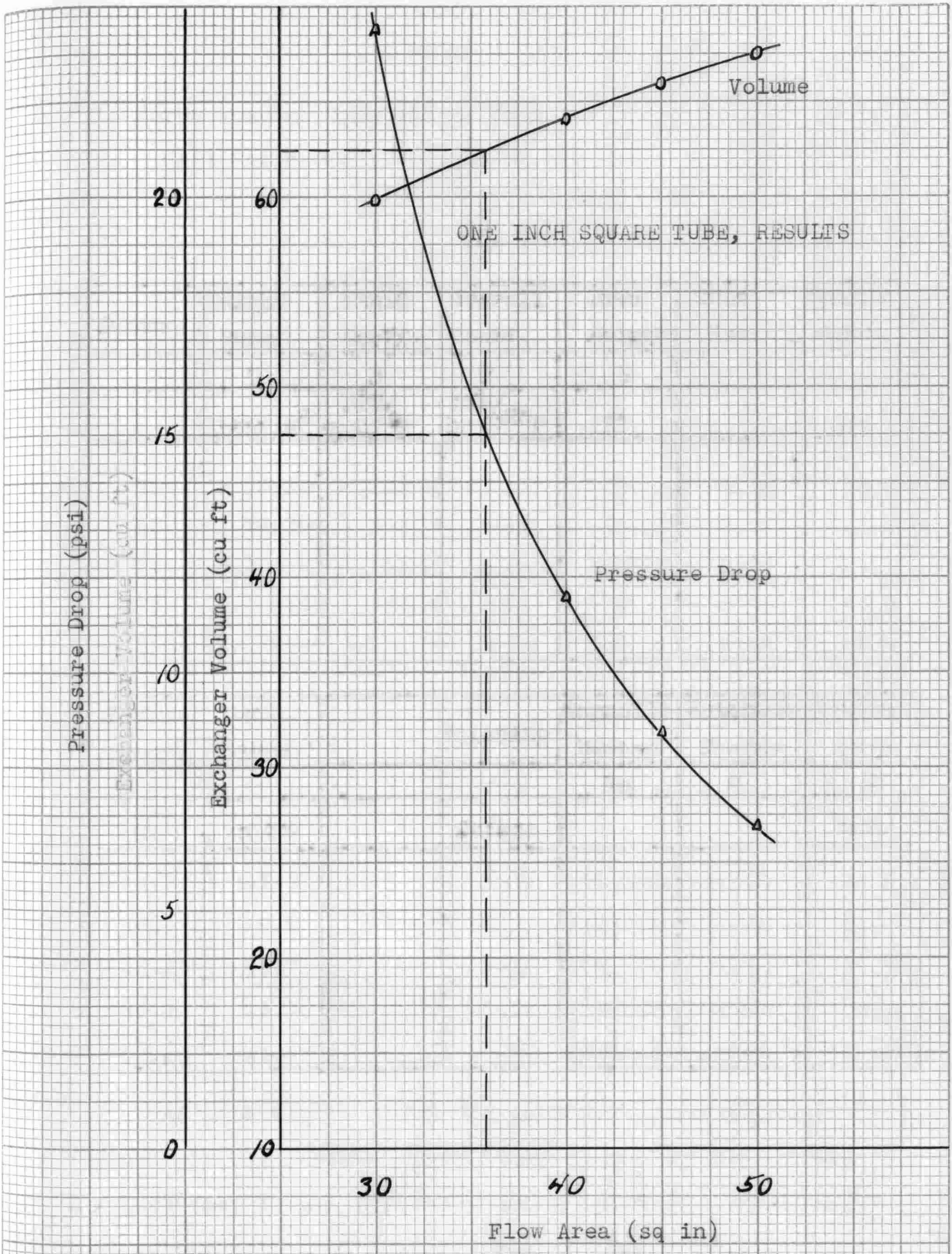
1. One Inch Square Tube

Wall thickness 0.0293 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 °F	BTU/hr ft^2 °F	ft		in
40	2.45×10^4	1.09×10^4	1560	96.0	10.25	57.5
50	1.95×10^4	0.97×10^4	1490	80.6	9.25	51.9
45	2.18×10^4	1.03×10^4	1530	87.1	9.75	54.1
30	3.26×10^4	1.23×10^4	1610	125.0	12.07	63.5

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
42.7	64.2		17.8	343,000	.00359	11.6
53.4	67.4		14.2	274,000	.00376	6.8
48.1	66.0		15.8	304,000	.00368	8.8
32.0	59.9		23.7	456,000	.00340	23.6

Figure 5, which is a graph of the above results, shows that the limiting factor in determining the exchanger volume is the allowable pressure drop. The optimum values are as follows; 15 psi pressure drop, 62.5 cu ft volume, and 36 tubes.



ONE INCH SQUARE TUBE, RESULTS
Volume
Pressure Drop
Pressure Drop (psi)
Exchanger Volume (cu ft)
Flow Area (sq in)
0
5
10
15
20
10
20
30
40
50
60
30
40
50

Figure 5

2. Three Quarter Inch Square Tube

Wall thickness of 0.0222 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in²	BTU/ hr ft² °F	BTU/ hr ft² °F	ft		in
70	2.48×10^4	1.28×10^4	1930	59.2	8.28	39.7
60	2.89×10^4	1.36×10^4	1990	67.0	8.99	41.9
55	3.15×10^4	1.43×10^4	2010	72.1	9.45	43.4
75	2.32×10^4	1.25×10^4	1920	55.9	7.96	38.7
65	2.67×10^4	1.33×10^4	1970	62.6	8.60	40.6

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
56.3	42.9		18.1	262,000	.00377	10.9
48.3	40.6		21.1	304,000	.00368	15.6
44.2	39.7		23.0	334.000	.00361	19.3
60.3	43.9		16.9	244,000	.00387	9.3
52.2	41.6		19.5	282,000	.00375	13.0

As in the case of the one inch tube, the limiting factor is the pressure drop. From figure 6, the optimum values are; 15 psi pressure drop, 40.9 cu ft volume, and 61 tubes.

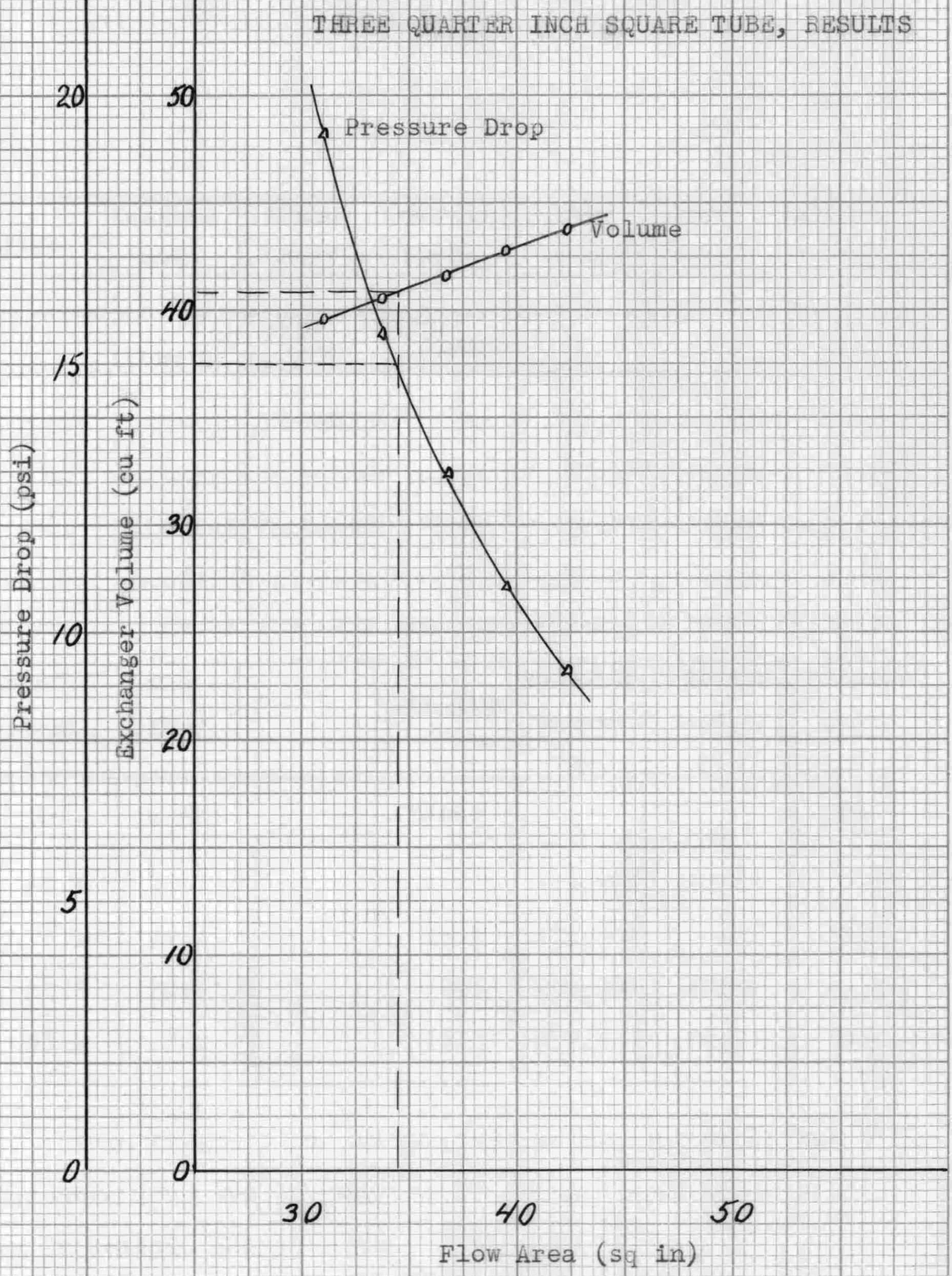

THREE QUARTER INCH SQUARE TUBE, RESULTS
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
30
40
50
Flow Area (sq in)

Figure 6

3. 0.6 Inch Square Tube

Wall thickness of 0.0177 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/ hr ft^2 °F	BTU/ hr ft^2 °F	ft		in
139	2.78×10^4	1.53×10^4	2380	30.2	5.50	27.6
116	3.30×10^4	1.66×10^4	2430	35.3	6.15	29.2
94	4.09×10^4	1.83×10^4	2510	42.5	7.16	31.9
84	4.65×10^4	1.94×10^4	2540	46.8	7.47	32.7

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
89.5	35.1		14.1	162,000	.00417	5.3
75.1	32.4		16.6	193,000	.00403	8.0
60.8	30.9		20.6	238,000	.00386	13.4
53.6	28.4		23.3	270,000	.00376	18.0

From figure 7 the optimum values as limited by the pressure drop are as follows; 15 psi pressure drop, 29.8 cu ft volume, and 90 tubes.

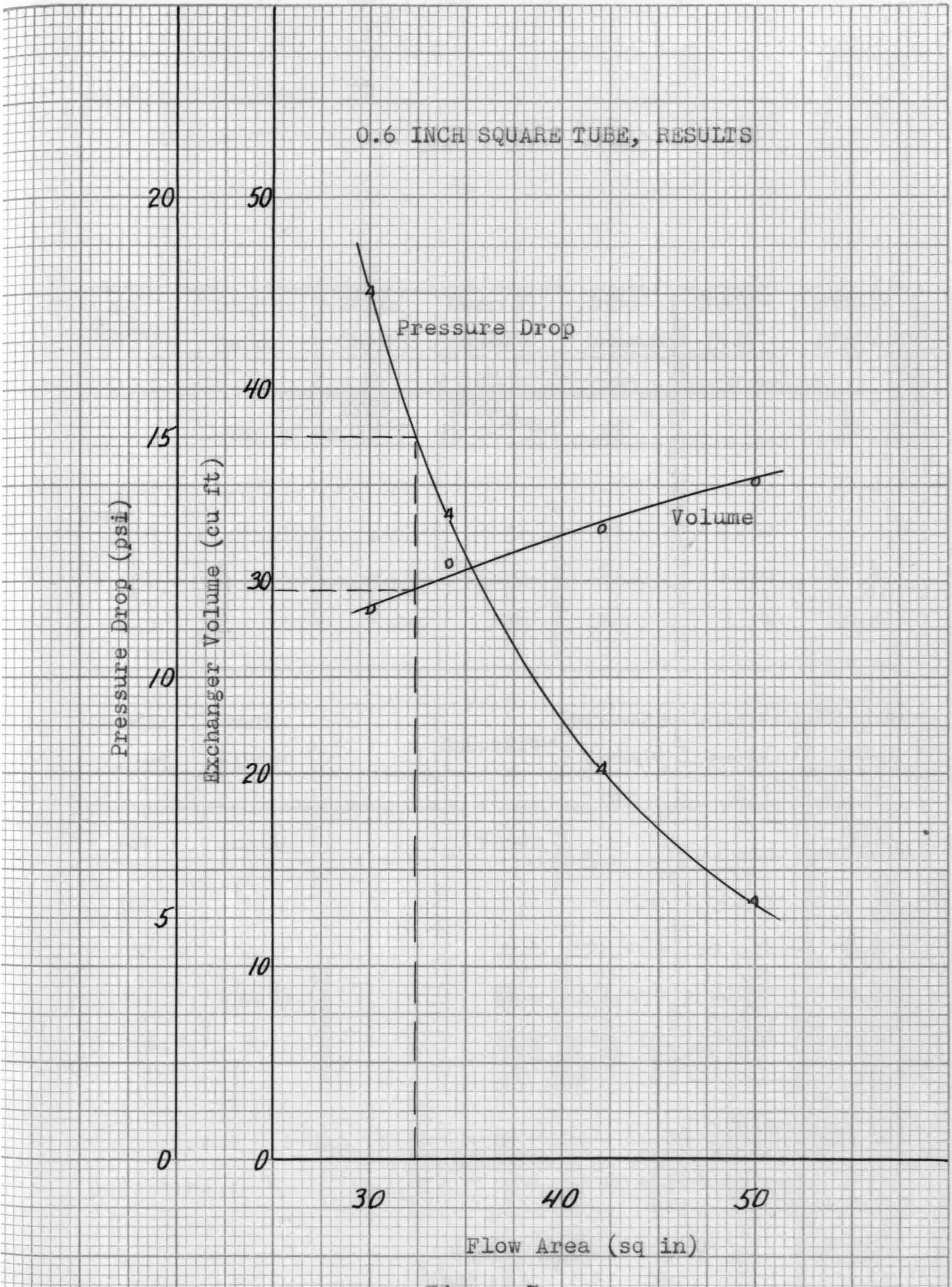

0.6 INCH SQUARE TUBE, RESULTS
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
30
40
50
Flow Area (sq in)

Figure 7

4. 0.5 Inch Square Tube

Wall thickness of 0.0148 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2°F	BTU/hr ft^2°F	ft		in
200	1.96×10^4	1.52×10^4	2670	22.4	4.55	23.2
160	2.45×10^4	1.62×10^4	2730	27.5	5.32	25.0
120	3.26×10^4	1.86×10^4	2860	35.0	6.39	27.3
140	2.80×10^4	1.72×10^4	2790	30.7	5.79	26.0

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
107.9	31.15		14.1	136,500	.00432	5.1
86.6	24.8		17.7	171,000	.00412	8.8
64.8	24.8		23.6	227,000	.00390	17.5
75.7	26.5		20.1	195,000	.00402	11.9

Figure 8 shows the optimum values to be as follows; 15 psi pressure drop, 25.3 cu ft volume, and 127 tubes.

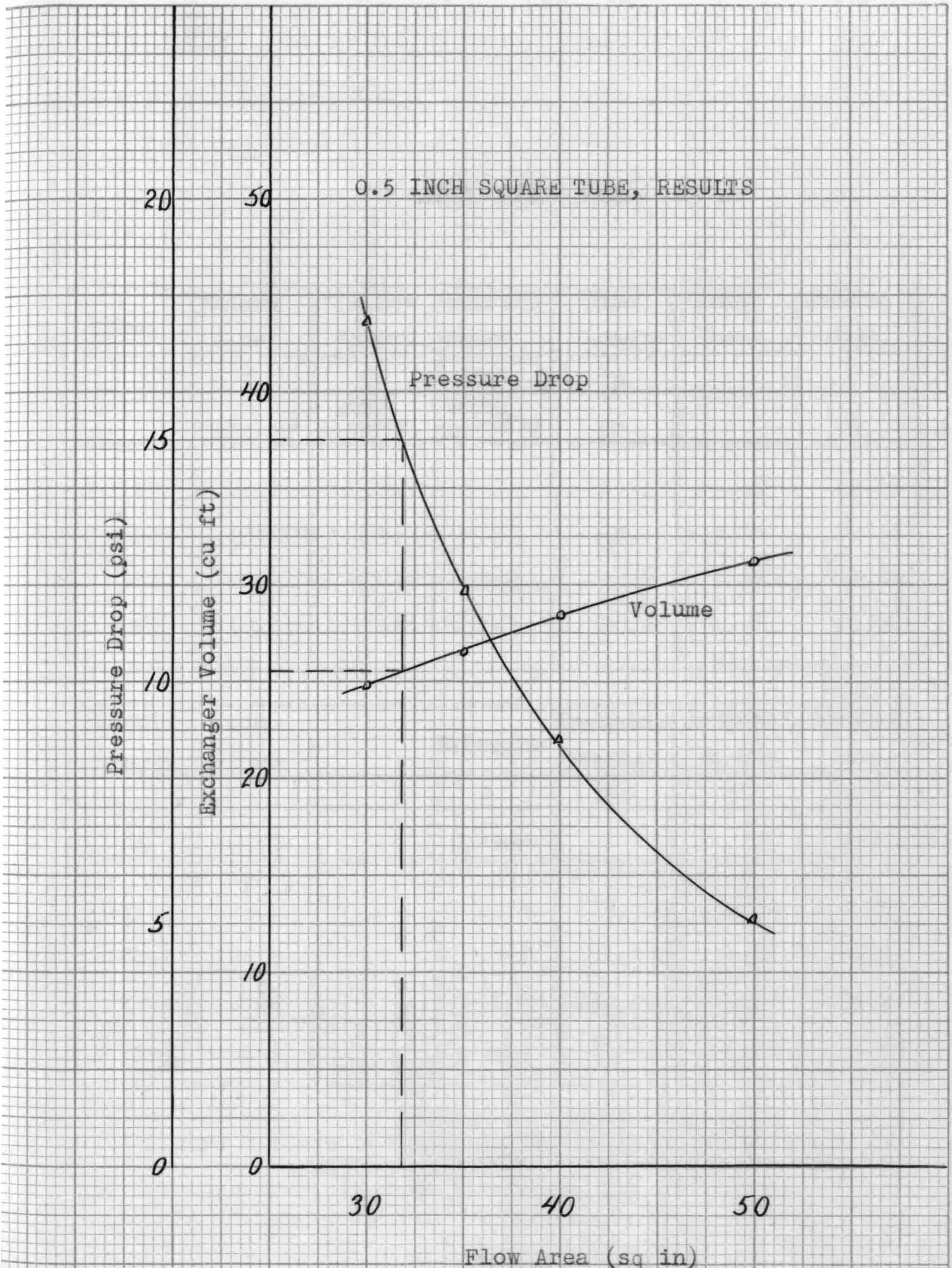

0.5 INCH SQUARE TUBE, RESULTS
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
Flow Area (sq in)
20
15
10
5
0
50
40
30
20
10
0
30
40
50

Figure 8

5. 0.4 Inch Square Tube

Tube wall thickness of 0.012 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in²	BTU/hr ft² °F	BTU/hr ft² °F	ft		in
313	1.96×10^4	1.70×10^4	3140	15.3	3.44	19.55
250	2.45×10^4	1.88×10^4	3260	18.4	3.98	20.48
187	3.26×10^4	2.07×10^4	3360	23.9	4.98	22.24
219	2.80×10^4	1.95×10^4	3290	20.8	4.45	21.36

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
135.6	29.6		14.15	108,800	.00454	4.85
108.2	25.5		17.7	136,000	.00432	8.16
81.1	21.9		23.7	182,000	.00407	16.45
95.0	24.0		20.2	155,000	.00420	11.88

Figure 9 shows the optimum values to be as follows; 15 psi pressure drop, 22.5 cu ft volume, and 197 tubes.

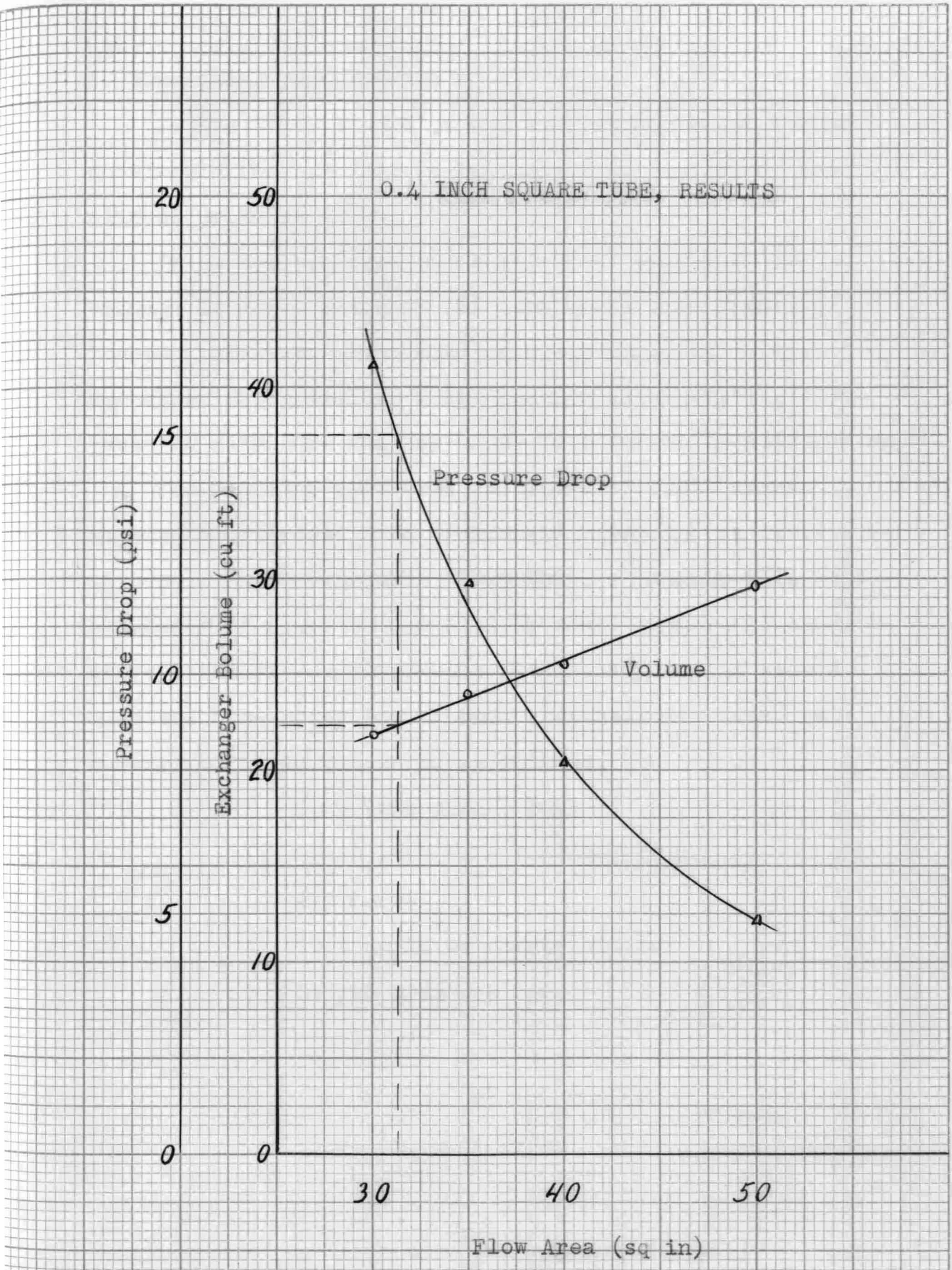

0.4 INCH SQUARE TUBE, RESULTS
Pressure Drop (psi)
Exchanger Bolume (cu ft)
Pressure Drop
Volume
Flow Area (sq in)
0
5
10
15
20
0
10
20
30
40
50
30
40
50

Figure 9

6. Summation, Square Tubes

Figure 10 is a plot of the minimum exchanger volume obtainable for the various tube sizes as limited by the allowed pressure drop.

It is obvious from the curves that even smaller volumes could be obtained for tube sizes less than 0.4 inches. However, the number of tubes required in the exchanger using smaller tubes and the manufacturing difficulties eliminate smaller tubes from further consideration.

The required tube wall thickness of 0.012 inches although sufficient from a strength viewpoint is difficult to handle in manufacture. Therefore, a minimum tube wall thickness of 0.020 inches will be used. Figure 11 shows the optimum values obtainable for a final design should a square tube prove better than a rectangular tube. A summary of these values is as follows;

0.40 inch square tube

0,020 inch tube wall thickness

213 tubes

29.4 cu ft exchanger volume

15 psi pressure drop

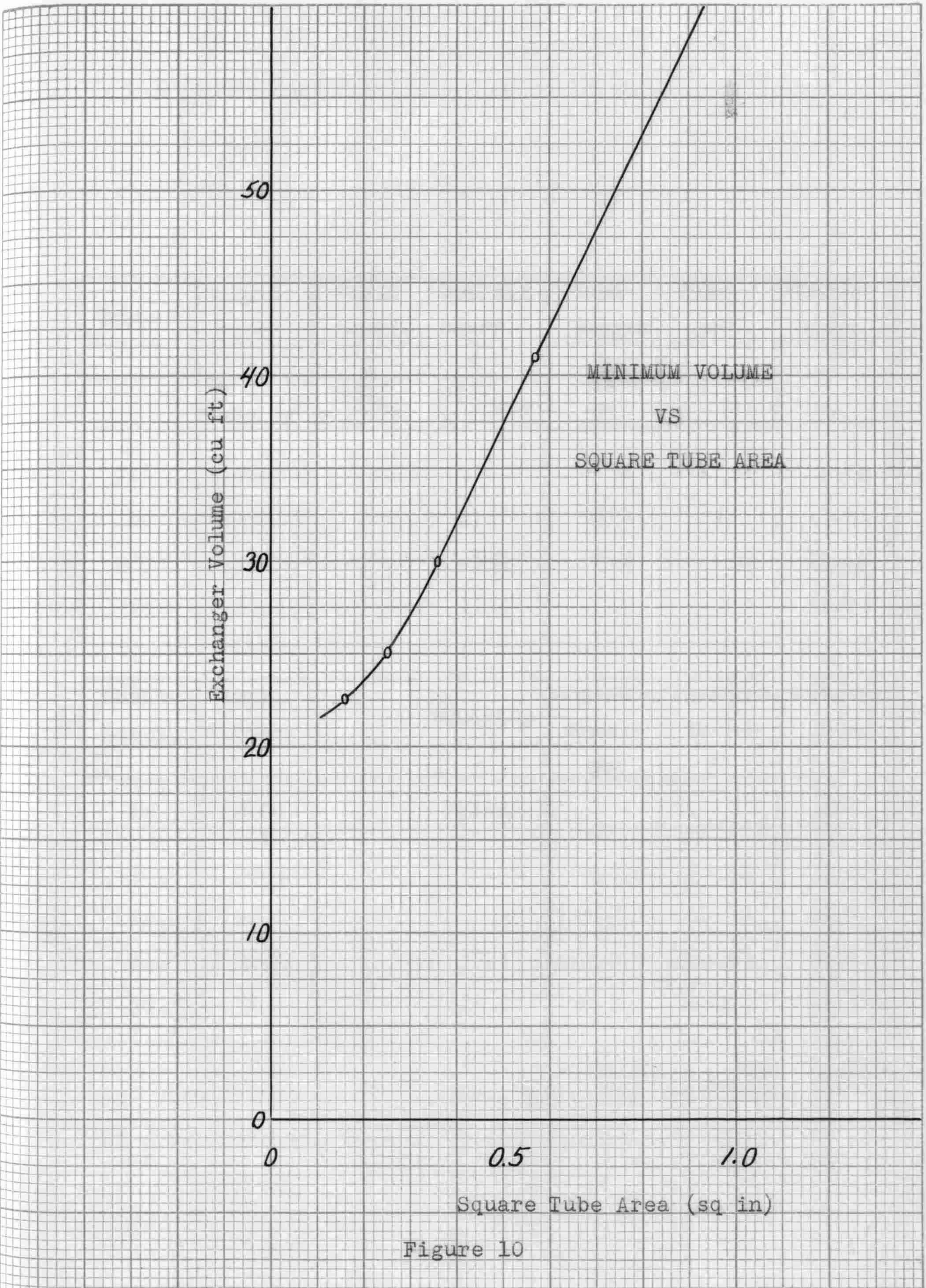

MINIMUM VOLUME
VS
SQUARE TUBE AREA
Exchanger Volume (cu ft)
50
40
30
20
10
0
0
0.5
1.0
Square Tube Area (sq in)

Figure 10

7. 0.4 Inch Square Tube With 0.020 Inch Wall

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2°F	BTU/hr ft^2°F	ft		in
313	1.96×10^4	1.70×10^4	2260	21.2	4.50	21.66
250	2.45×10^4	1.88×10^4	2320	25.8	5.25	23.01
187	3.26×10^4	2.07×10^4	2380	33.8	6.50	25.26
219	2.80×10^4	1.95×10^4	2350	29.2	5.79	23.98

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
140.6	36.4		14.15	108,800	.00454	4.8
112.3	32.0		17.7	136,000	.00432	9.9
84.5	28.2		23.7	182,000	.00407	20.4
98.5	30.1		20.2	155,000	.00420	13.7

From figure 11 the optimum values are as follows; 15 psi pressure drop, 29.4 cu ft volume, and 213 tubes.

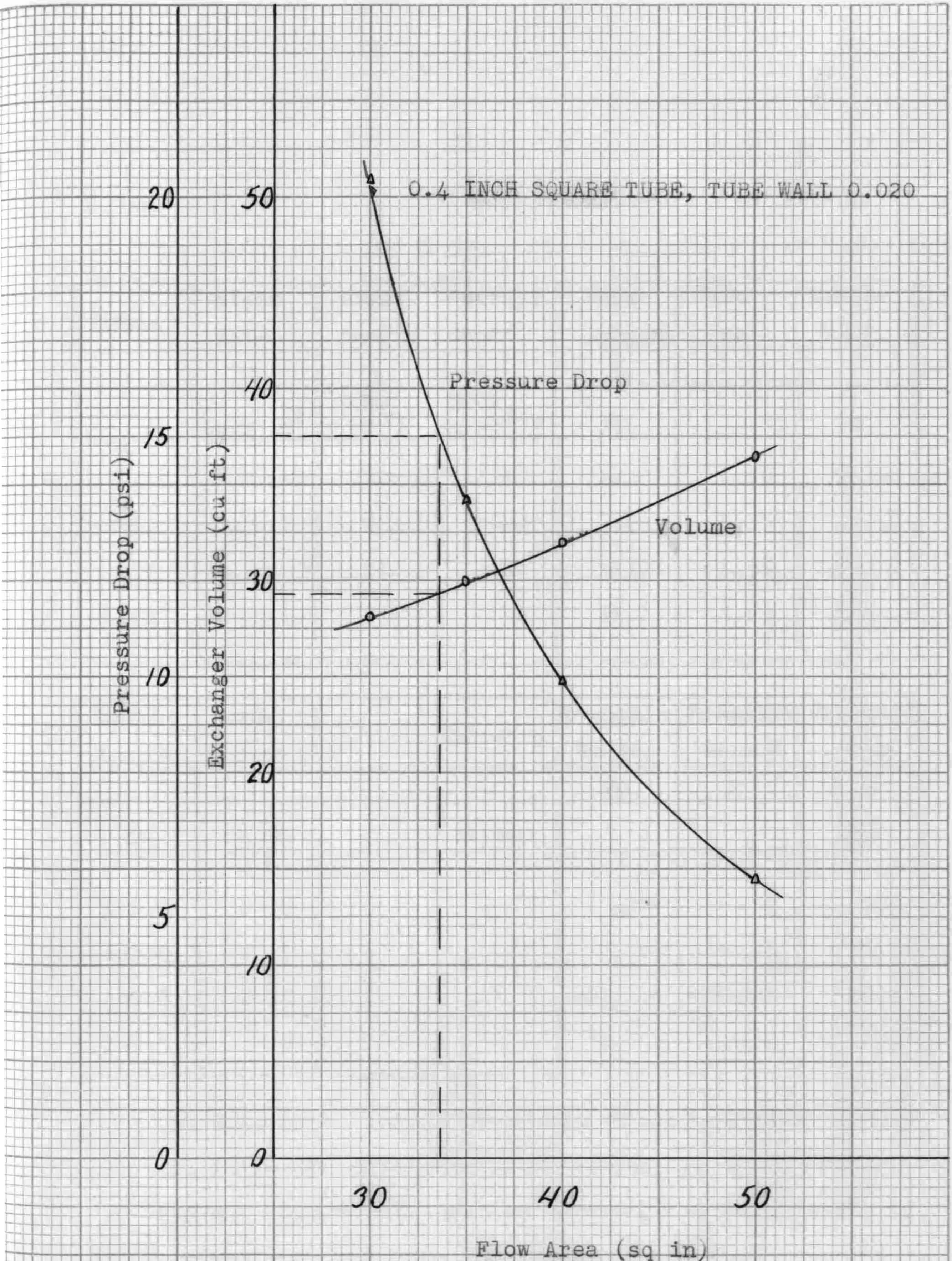

0.4 INCH SQUARE TUBE, TUBE WALL 0.020
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
30
40
50
Flow Area (sq in)

Figure 11

CHAPTER VIII

RECTANGULAR TUBES

1. b/a 1.2

Tube wall of 0.020 inches, hydraulic diameter of 0.397 in

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 $^\circ$F	BTU/hr ft^2 $^\circ$F	ft		in
313	1.96×10^4	1.72×10^4	2290	20.8	4.50	20.86
250	2.45×10^4	1.90×10^4	2340	25.5	5.35	22.21
187	3.26×10^4	2.11×10^4	2400	33.2	6.58	24.22
219	2.80×10^4	1.99×10^4	2370	28.7	5.85	23.04

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
156.6	37.9		14.15	108,000	.00454	5.8
125.0	33.6		17.7	135,000	.00432	9.9
93.6	29.1		23.7	180,500	.00407	20.4
109.5	31.3		20.2	154,000	.00420	13.7

b, Tube width (longitudinal direction) is 0.451 inches

a, Tube thickness (radial direction) is 0.355 inches

Figure 12 shows the optimum values to be as follows; 15 psi pressure drop, 30.6 cu ft volume, and 210 tubes.

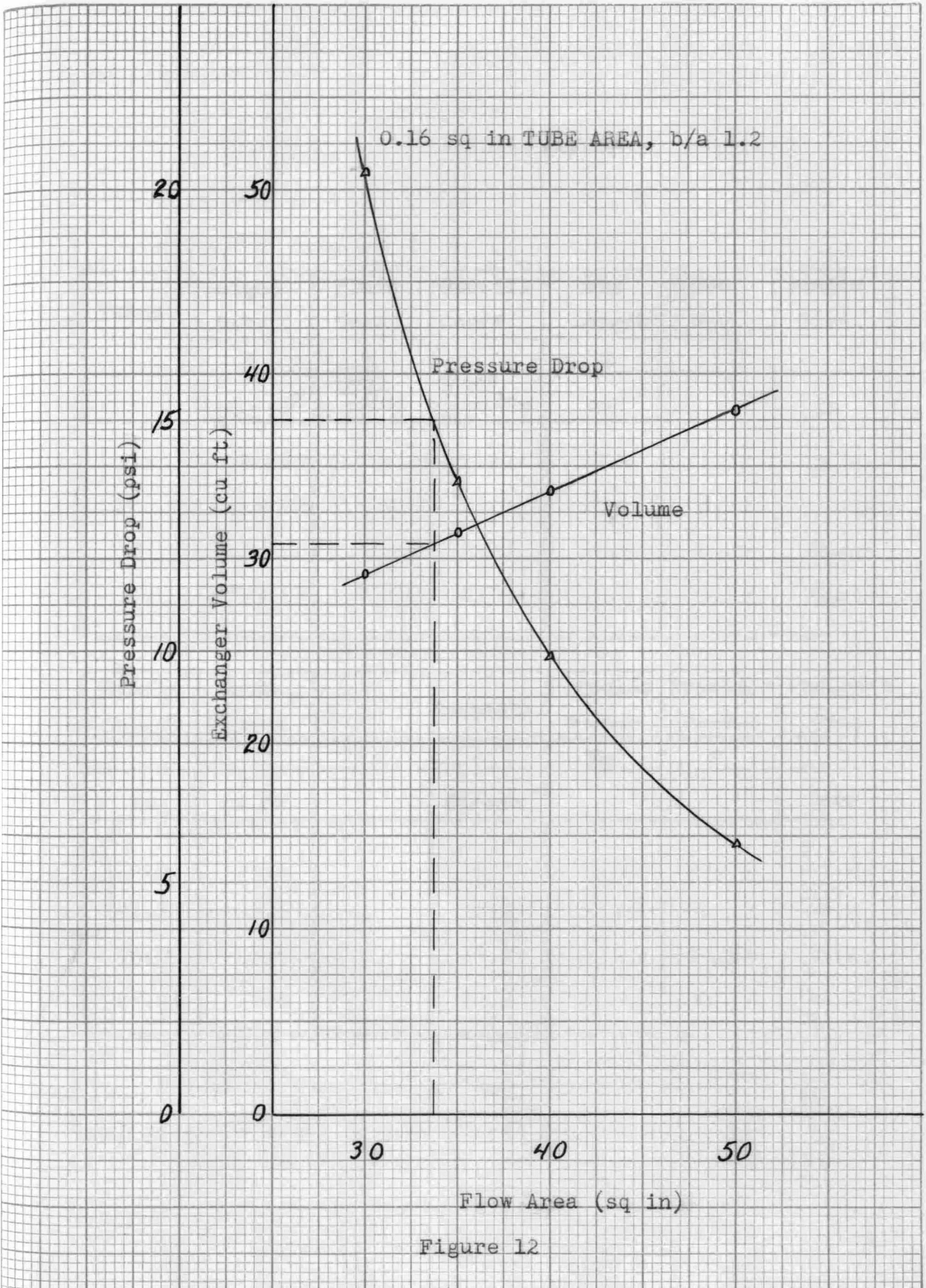

0.16 sq in TUBE AREA, b/a 1.2
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
0
5
10
15
20
0
10
20
30
40
50
30
40
50
Flow Area (sq in)

Figure 12

2. a/b 1.2

Tube width is 0.355 inches

Tube thickness is 0.451 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.397 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 °F	BTU/hr ft^2 °F	ft		in
313	1.96x10^4	1.72x10^4	2290	20.8	4.33	22.20
250	2.45x10^4	1.92x10^4	2340	25.5	5.10	23.73
187	3.26x10^4	2.11x10^4	2400	33.2	6.26	26.06
219	2.80x10^4	1.99x10^4	2370	28.7	5.59	24.70

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
126.6	34.1		14.15	108,000	.00454	5.8
101.3	30.4		17.7	135,000	.00432	9.9
76.0	26.8		23.7	180.500	.00407	20.4
88.7	29.0		20.2	154,000	.00420	13.7

Figure 13 shows the optimum values to be as follows; 15 psi pressure drop, 28.3 cu ft volume, and 211 tubes.

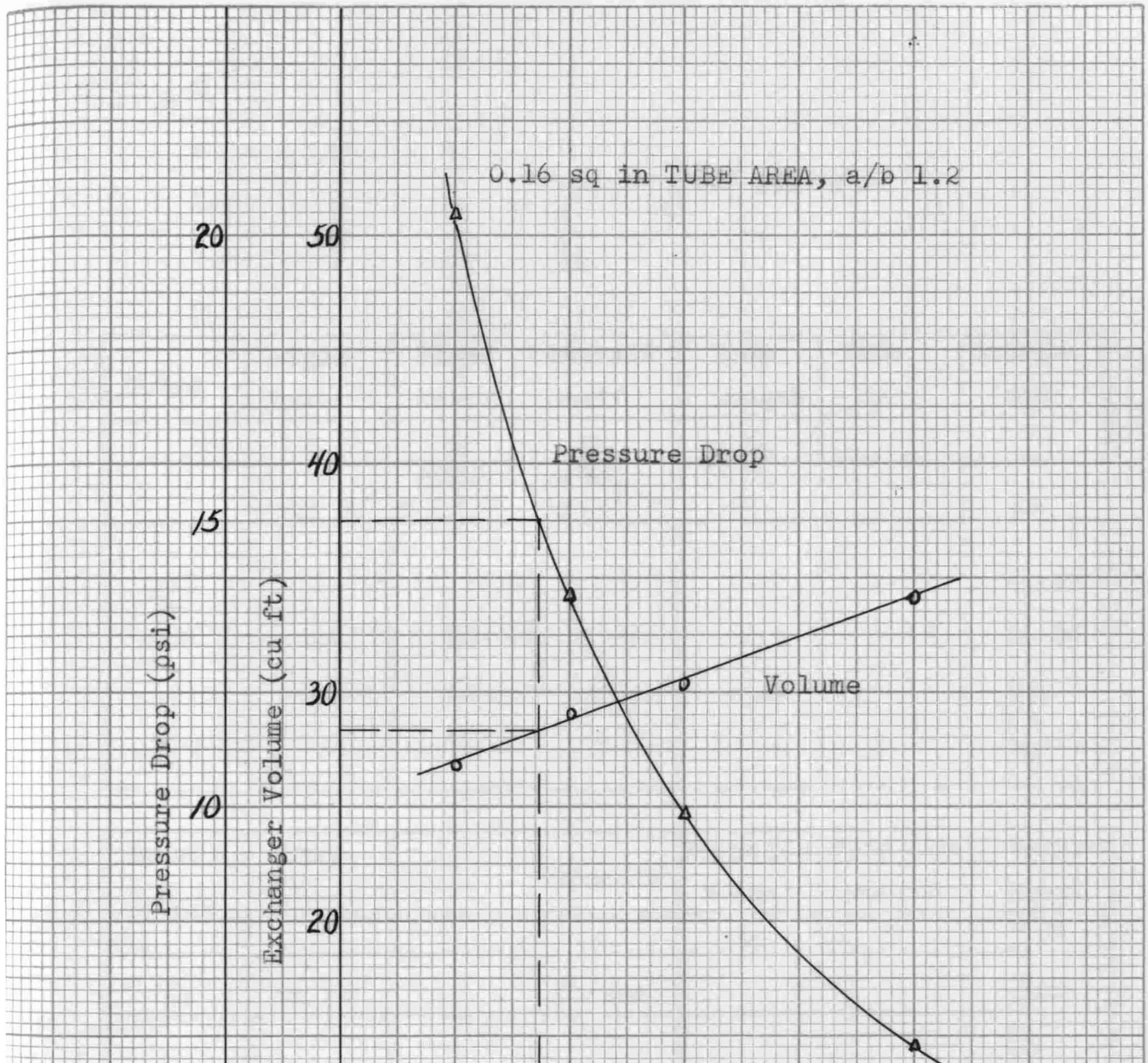

0.16 sq in TUBE AREA, a/b 1.2
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
30
40
50
Flow Area (sq in)

Figure 13

3. a/b 1.4

Tube width, b, is 0.338 inches

Tube thickness, a, is 0.474 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.394 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/ hr ft^2 °F	BTU/ hr ft^2 °F	ft		in
313	1.96x10^4	1.75x10^4	2290	20.6	4.25	22.40
250	2.45x10^4	1.90x10^4	2330	25.3	5.17	24.32
187	3.26x10^4	2.11x10^4	2390	33.0	6.15	26.37
219	2.80x10^4	2.01x10^4	2360	28.6	5.47	24.95

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
121.4	33.1		14.15	167,600	.00454	5.78
97.1	28.3		17.7	134,500	.00432	9.87
72.8	25.9		23.7	180,000	.00407	20.30
85.0	27.8		20.2	153,500	.00420	13.70

Figure 14 shows the optimum values to be as follows; 15 psi pressure drop, 26.8 cu ft bolume, and 209 tubes.

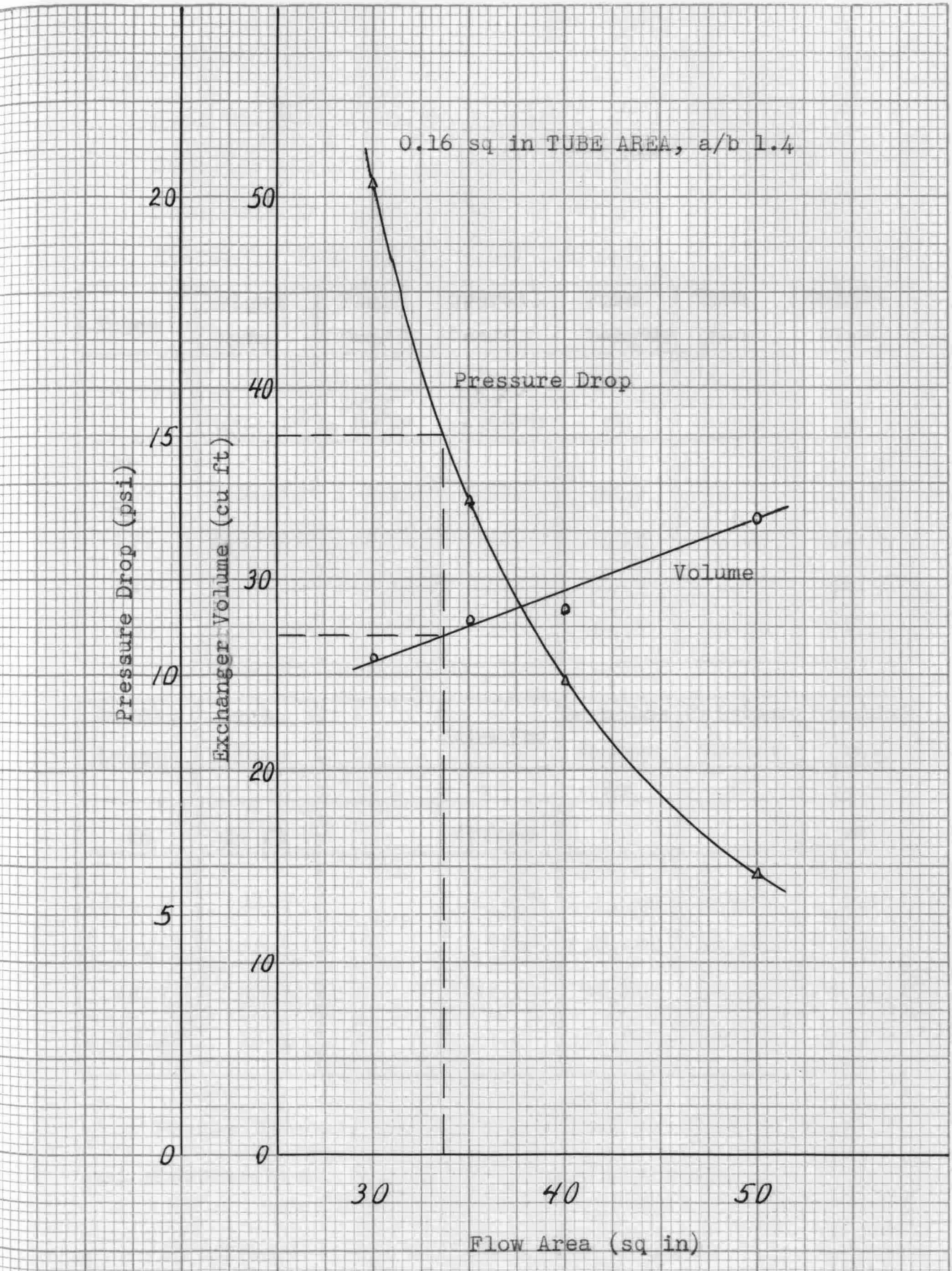

0.16 sq in TUBE AREA, a/b 1.4
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
0
5
10
15
20
0
10
20
30
40
50
30
40
50
Flow Area (sq in)

Figure 14

4. a/b 1.6

Tube width, b, is 0.316 inches

Tube thickness, a, is 0.507 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.389 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 °F	BTU/hr ft^2 °F	ft		in
313	1.96×10^4	1.77×10^4	2300	20.3	4.25	22.71
250	2.45×10^4	1.90×10^4	2345	24.9	4.89	24.34
187	3.26×10^4	2.11×10^4	2400	32.5	5.99	26.80
219	2.80×10^4	2.00×10^4	2370	28.1	5.20	25.02

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	p
in	cu ft		ft/sec			psi
114.5	31.9		14.15	106,000	.00455	5.76
91.5	28.5		17.7	132,000	.00436	9.97
68.7	25.5		23.7	176,000	.00411	20.40
80.2	26.4		20.2	151,000	.00424	13,75

Figure 15 shows the optimum values to be as follows; 15 psi pressure drop, 26.4 cu ft volume, 213 tubes.

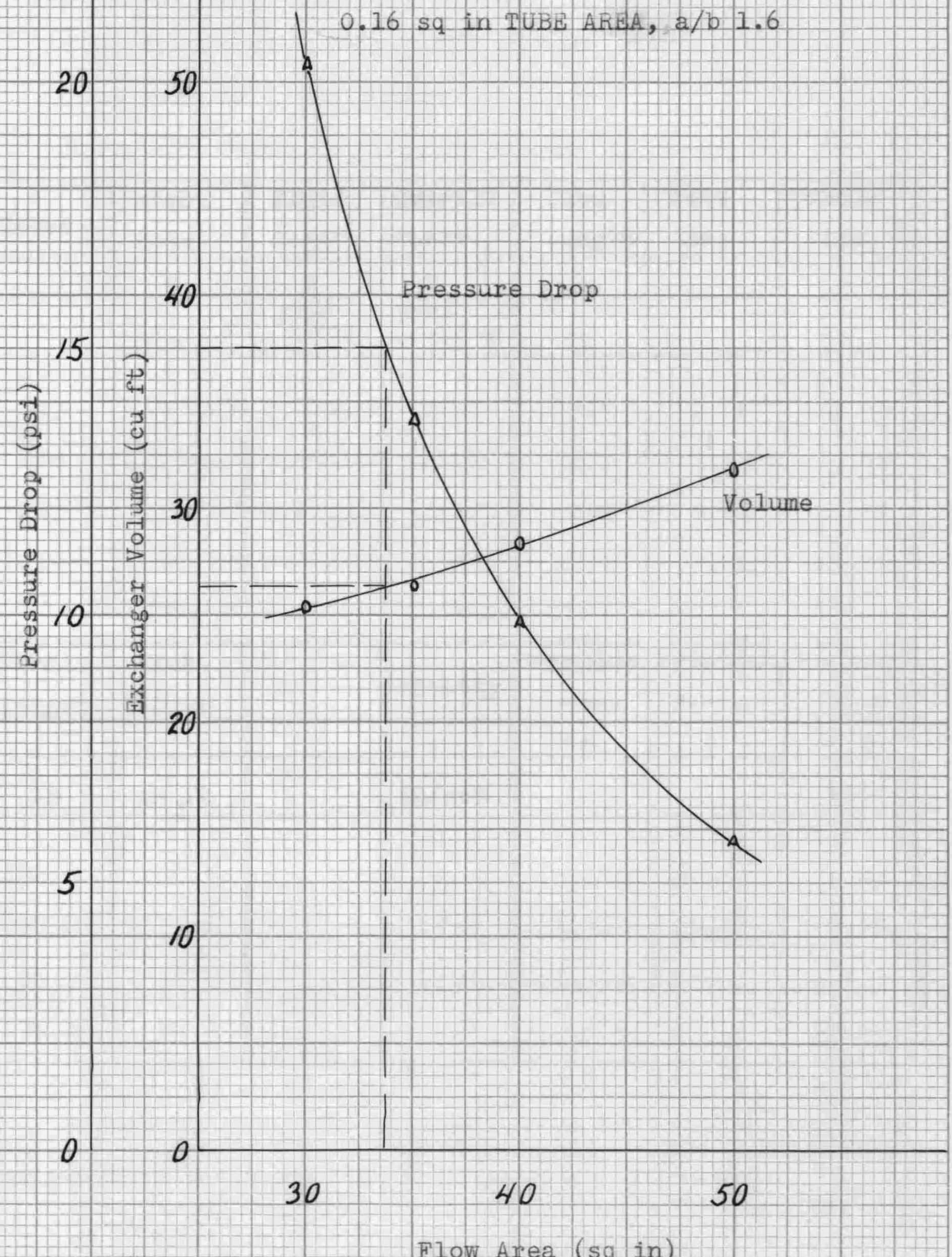

0.16 sq in TUBE AREA, a/b 1.6
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
Pressure Drop
Volume
30
40
50
Flow Area (sq in)

Figure 15

5. a/b 1.8

Tube width, b, is 0.298 inches

Tube thickness, a, is 0.537 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.383 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 °F	BTU/hr ft^2 °F	ft		in
313	1.96×10^4	1.77×10^4	2300	19.9	4.06	22.90
250	2.45×10^4	1.90×10^4	2340	24.5	4.74	24.62
187	3.26×10^4	2.11×10^4	2400	32.0	5.82	27.12
219	2.80×10^4	2.00×10^4	2370	27.7	5.22	25.72

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
108.8	30.7		14.15	105,500	.00455	5.75
87.0	27.8		17.7	131,500	.00436	9.96
65.3	24.7		23.7	175,000	.00411	20.35
76.1	26.2		20.2	150,000	.00426	13.70

Figure 16 shows the optimum values to be as follows; 15 psi pressure drop, 25.7 cu ft volume, and 212 tubes.

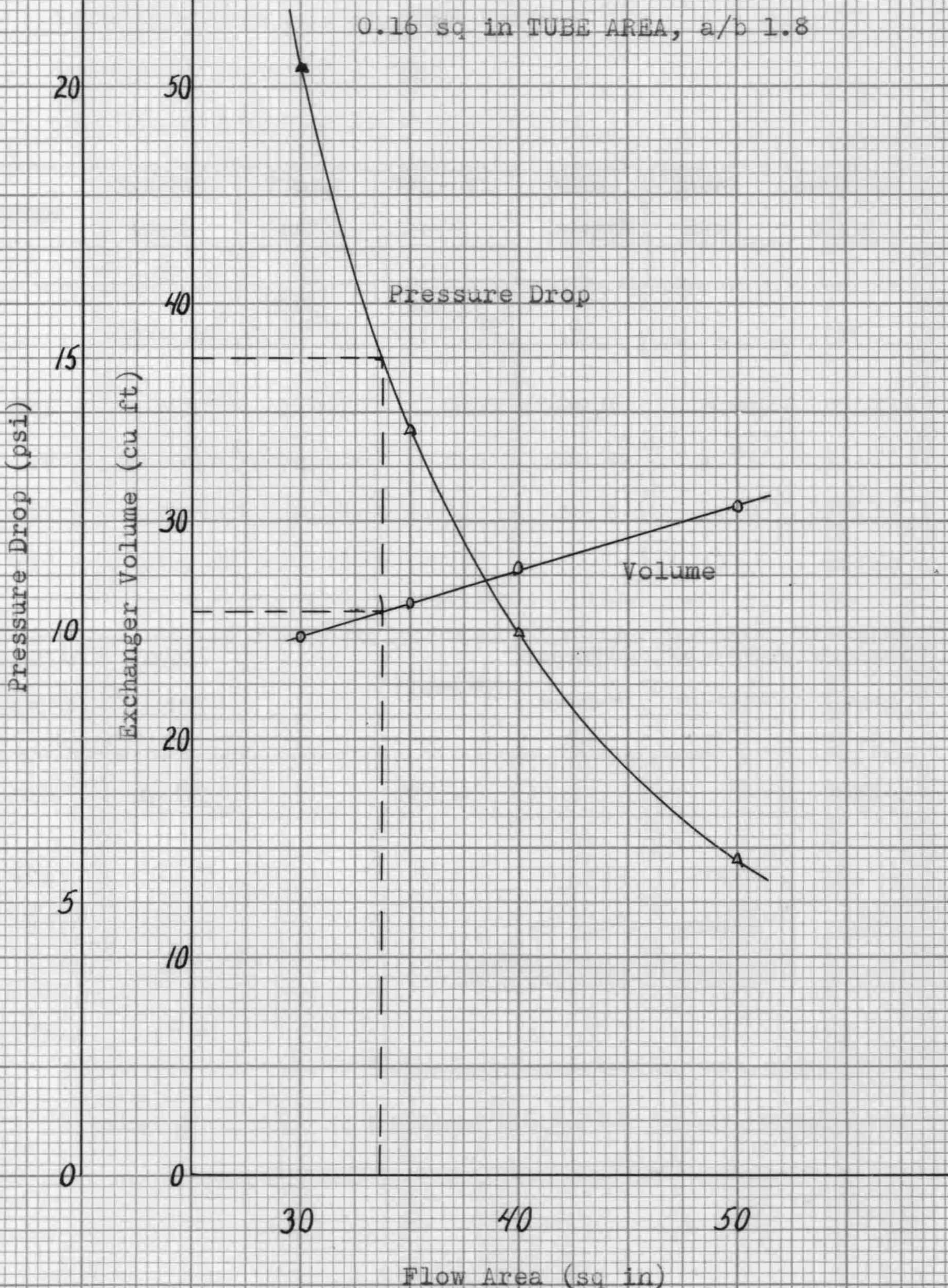

0.16 sq in TUBE AREA, a/b 1.8
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
Flow Area (sq in)
0
5
10
15
20
0
10
20
30
40
50
30
40
50

Figure 16

6. a/b 2.0

Tube width, b, is 0.283 inches

Tube thickness, a, is 0.565 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.377 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/hr ft^2 °F	BTU/hr ft^2 °F	ft		in
313	1.96×10^4	1.78×10^4	2310	19.6	3.96	23.13
250	2.45×10^4	1.94×10^4	2350	24.1	4.74	25.01
187	3.26×10^4	2.12×10^4	2410	31.4	5.70	27.39
219	2.80×10^4	2.01×10^4	2380	27.1	5.10	25.94

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
104.2	29.9		14.15	105,000	.00456	5.74
83.5	27.4		17.7	131,000	.00436	9.95
62.7	24.1		23.7	175,000	.00411	20.3
73.2	25.6		20.2	150,000	.00426	13.7

Figure 17 shows the optimum values to be as follows; 15 psi pressure drop, 25.3 cu ft volume, and 211 tubes.

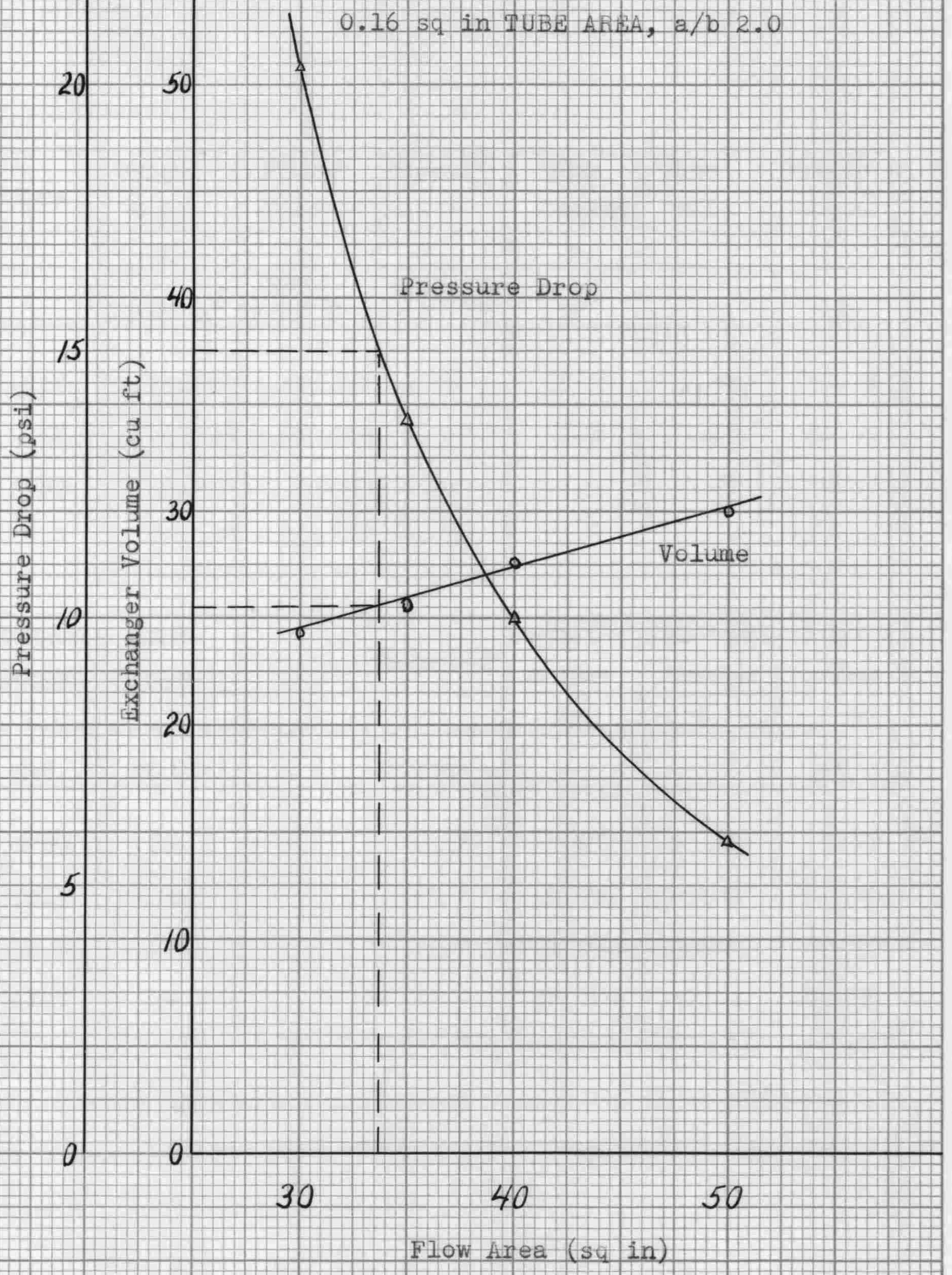

0.16 sq in TUBE AREA, a/b 2.0
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
Flow Area (sq in)
0
5
10
15
20
0
10
20
30
40
50
30
40
50

Figure 17

7. a/b 3.0

Tube width, b, is 0.231 inches

Tube thickness, a, is 0.593 inches

Tube wall is 0.020 inches

Hydraulic diameter is 0.346 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in^2	BTU/ hr ft^2°F	BTU/ hr ft^2°F	ft		in
313	1.96×10^4	1.88×10^4	2340	17.75	3.52	23.73
250	2.45×10^4	2.02×10^4	2380	21.8	4.13	25.56
187	3.26×10^4	2.25×10^4	2430	28.6	5.07	28.33
219	2.80×10^4	2.11×10^4	2400	24.7	4.54	26.76

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
87.9	26.4		14.15	93,700	.00468	5.82
70.5	24.0		17.7	118,000	.00450	10.10
52.8	21.6		23.7	158,000	.00421	20.50
61.7	22.8		20.2	134,500	.00435	13.85

Figure 18 shows the optimum values to be as follows; 15 psi pressure drop, 22.5 cu ft volume, and 212 tubes.

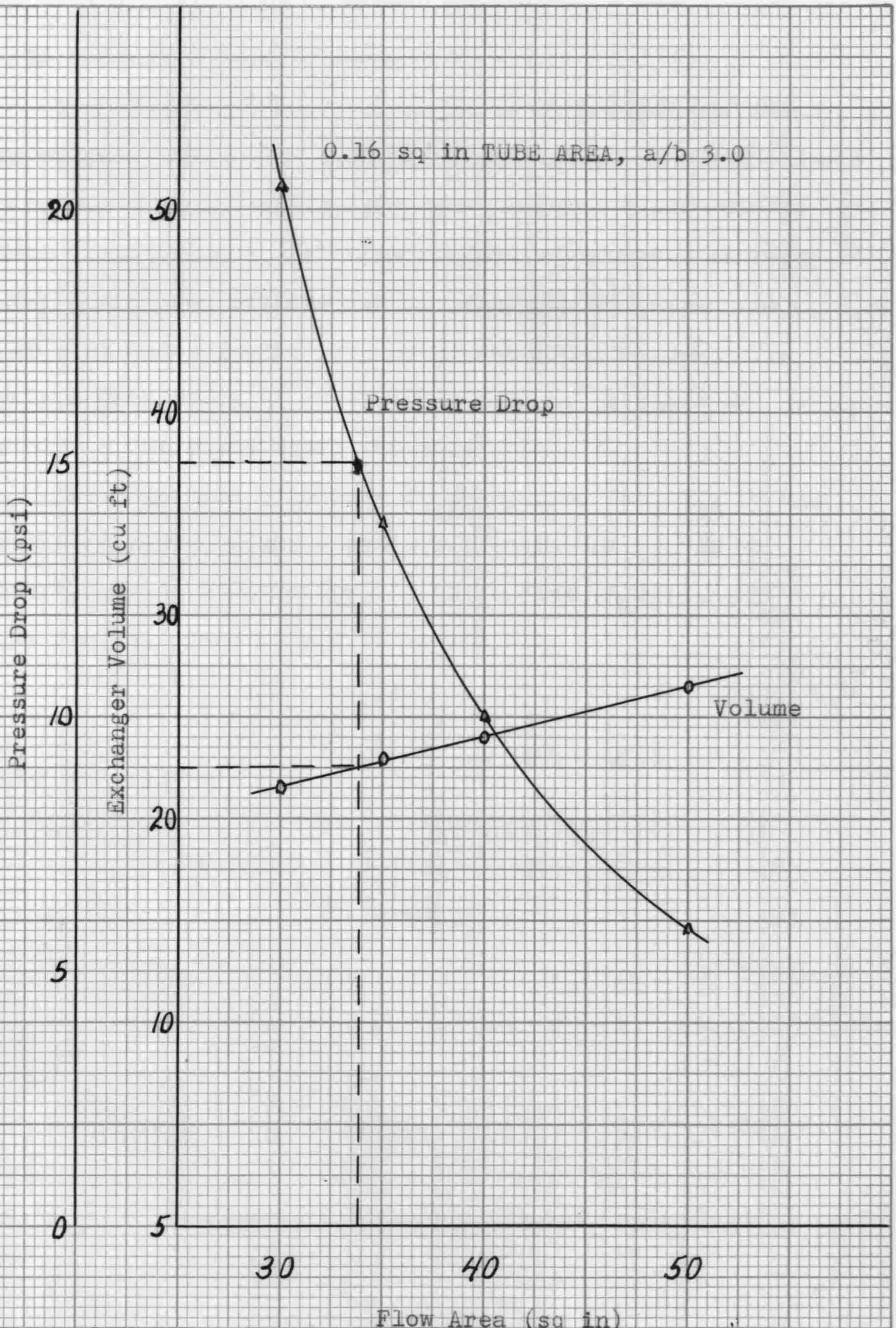
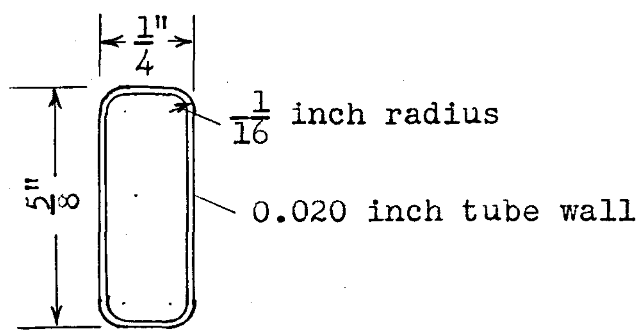

0.16 sq in TUBE AREA, a/b 3.0
Pressure Drop
Volume
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
5
30
40
50
Flow Area (sq in)

Figure 18

8. Rectangular Tubes, Summation

From figure 19 the conclusion is drawn that the minimum volume, limited by pressure drop, for a given tube area decreases with increasing thickness (radial direction)-width (longitudinal direction) ratio. Therefore, at this point sufficient information is available to choose the tube size and shape to be used in the final design. The optimum tube would be as small in area as possible and have as high a thickness-width ratio as possible.

The selection of a tube is determined not only by the desire to meet the above characteristics but by the feasibility of manufacture. A compromise selection of tube for the final design is as follows;



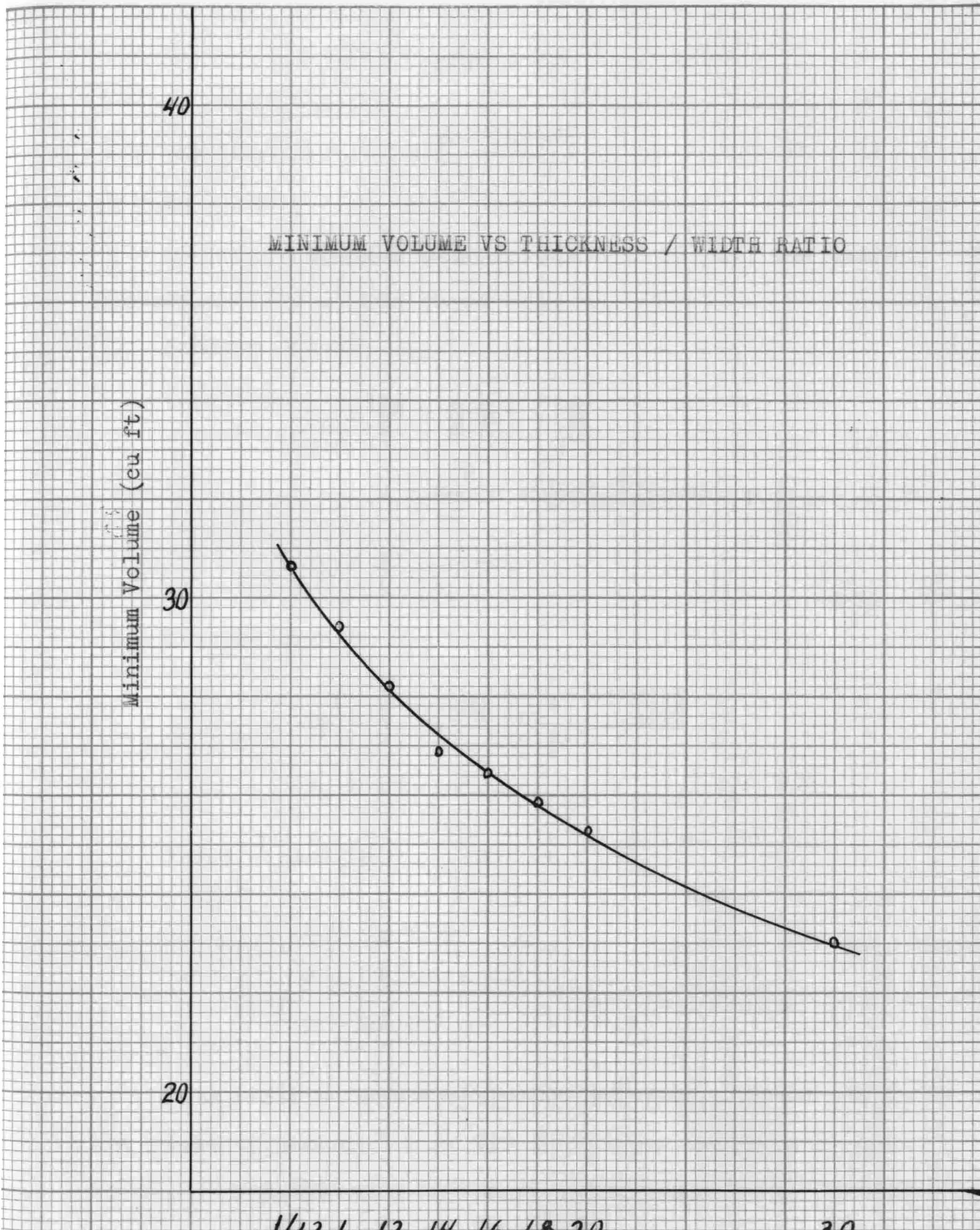

MINIMUM VOLUME VS THICKNESS / WIDTH RATIO
Minimum Volume (cu ft)
40
30
20
1/1.2
1
1.2
1.4
1.6
1.8
2.0
3.0
Thickness / Width Ratio

Figure 19

CHAPTER IX

FINAL DESIGN

1. Tube Loading

Under the maximum tube pressure of 50 psi, the maximum stress in the tube is as follows;

Bending stress, S_M,

$$S_M = \frac{Mc}{I} = \frac{M\ u/2}{u^3/12} = \frac{6M}{u^2}$$

$$M = \frac{wl^2}{12} = \frac{50(0.625-0.040)}{12} = 1.425 \text{ lb in}$$

$$S_M = 21{,}300 \text{ psi}$$

Tensile stress, S_T,

$$S_T = \frac{F}{u} = \frac{50\sqrt{0.585^2 + 0.210^2}}{2\text{x}0.020}$$

$$S_T = 726 \text{ psi}$$

Total stress, S,

$$S = S_M + S_T$$

$$S = 21{,}300 + 726$$

$$S = 22{,}000 \text{ psi}$$

This maximum stress of 22,000 psi when compared to the yield strength of type 347 stainless, which is 35,000 to 40,000 psi, gives a factor of safety of from 1.6 to 1.8.

An additional factor of safety is provided in this design by the mutual support of the closely packed tubes.

2. Exchanger Volume

Area of tube is 0.122 sq in

Hydraulic diameter is 0.306 inches

Tubes	Weight Rate	Fluid Coeff.	Overall Coeff.	Tube Length	Tube Rev.	Bundle Diam.
	G	h	U	L		
	lb/hr in²	BTU/hr ft² °F	BTU/hr ft² °F	ft		in
350	2.26×10^4	2.13×10^4	2410	17.8	3.73	22.62
300	2.63×10^4	2.26×10^4	2440	20.5	4.08	23.74
250	3.16×10^4	2.42×10^4	2470	24.3	4.65	25.20
400	1.98×10^4	2.01×10^4	2380	16.75	3.31	21.76

Bundle Length	Exch. Volume		Velocity	Reynolds Number	Friction Factor	Pressure Drop
			v	Re	f	P
in	cu ft		ft/sec			psi
91.0	25.2		16.3	97,300	.00462	8.15
78.0	23.5		19.0	113,300	.00450	12.00
65.0	21.6		22.8	136,000	.00433	18.80
104.0	27.1		14.2	85,000	.00474	6.08

Figure 20 shows the approximate exchanger volume to be 22.3 cu ft and the pressure drop to be 15 psi for an exchanger of 270 tubes.

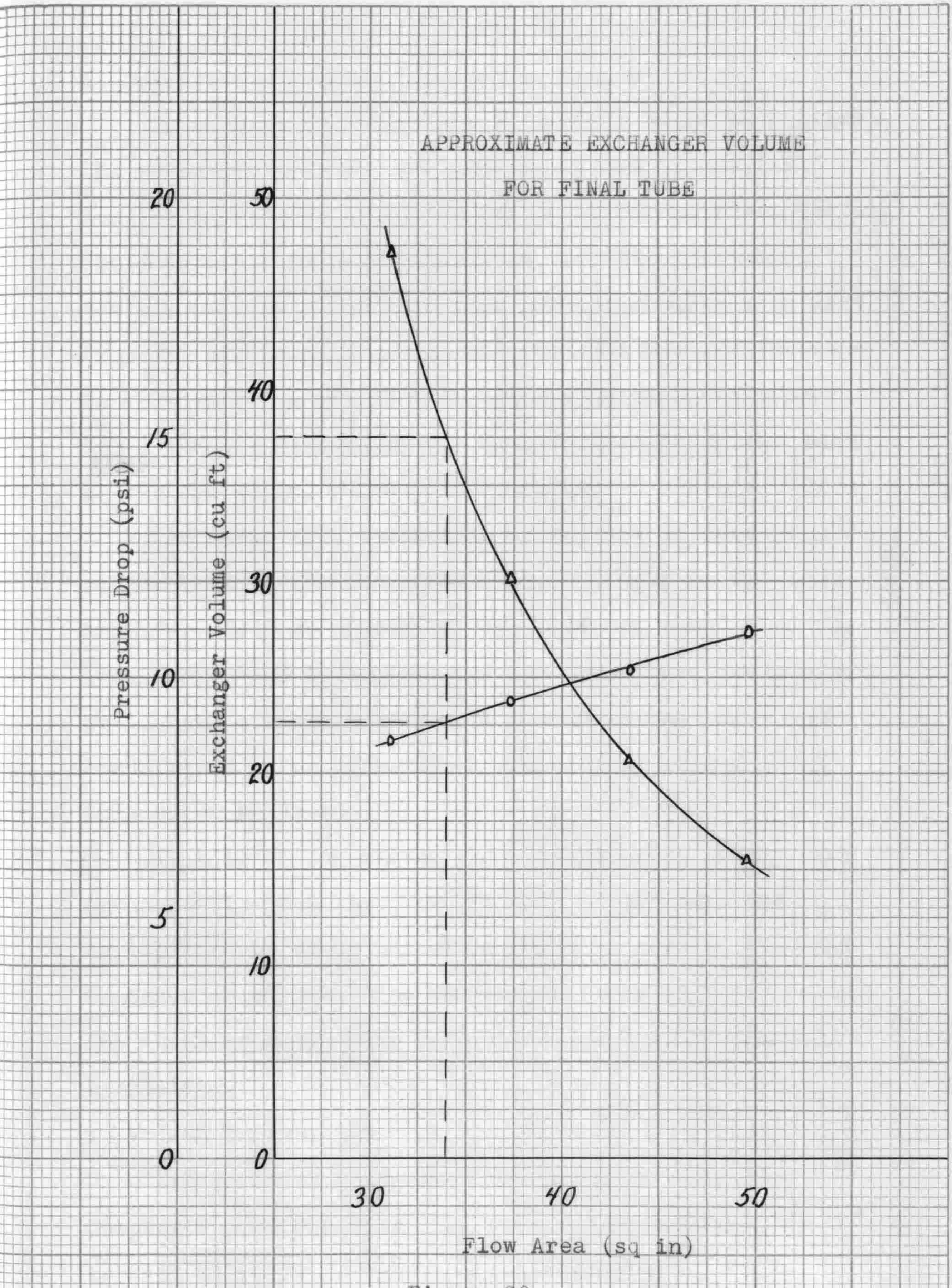

APPROXIMATE EXCHANGER VOLUME
FOR FINAL TUBE
Pressure Drop (psi)
Exchanger Volume (cu ft)
20
15
10
5
0
50
40
30
20
10
0
30
40
50
Flow Area (sq in)

Figure 20

3. Corrections to Heat Transfer Equations

To correct the calculations made in order to estimate the exchanger volume, the tube is divided into three lengths corresponding to the innermost one half revolution, the outermost one half revolution, and the remaining central portion. The equations used previously will hold true for the central portion but not for the end portions in that sections of the tube are not adjacent to other tubes but are bounded by the central headers or the shell of the exchanger.

The inner half revolution would have an area of 46.3 square inches affected. The maximum effect on the heat transfer would be 38,000 BTU/hr ($UA\Delta T$) or 0.039 % of the total, which can be neglected. The outer half revolution would have a maximum effect of 0.070% of the total heat transferred. The corrections can also be neglected for this outer portion.

In view of the above, the configuration of the heat exchanger is such that the possible corrections are negligible and none, therefore, need be made to the heat transfer equations. The only other corrections to be made are those of using more accurate values of the physical properties.

4. Volume Corrections

The physical properties of sodium as given in Chapter I give the following more accurate values to be used in correcting our calculations;

	Cooled	Heated
Density	55.0	55.5
Viscosity	0.85	0.91
Specific Heat	0.340	0.339
Thermal Conductivity	43.6	44.0

Exchanger Data

270 tubes

0.122 sq in tube area

1.590 inch tube circumference (inner)

0.020 inch tube wall

0.306 inch hydraulic diameter

70.1 inch bundle length

Cooled Fluid

Weight Rate, $W_h = 0.98\times10^6$ lb/hr

Volume Rate, $V_h = 17{,}800$ ft^3/hr

Weight Rate, $G = 2.97\times10^4$ lb/hr in^2

Fluid Coeff., $h = 2.39\times10^4$ BTU/hr $ft^{2}{}^{o}F$

Overall Coeff., $U = 2.47\times10^4$ BTU/hr $ft^{2}{}^{o}F$

Tube Length, L = 22.1 ft, 265 in

Heated Fluid

Weight Rate, $W_c = 0.982$ lb/hr

Volume Rate, $V_c = 17{,}670$ ft^3/hr

Weight Rate, $G = 2.98\times10^4$ lb/hr in^2

Fluid Coeff., $h = 2.40\times10^4$ BTU/hr $ft^{2}{}^{o}F$

Overall Coeff., $U = 2.47\times10^4$ BTU/hr $ft^{2}{}^{o}F$

Tube Length, L = 22.1 ft, 265 in

CHAPTER X

CONSTRUCTION

1. Central Headers and Tubes

Figure 21a shows the start of construction with the first section of the headers in place ready to receive the first row of tubes which are to be set in place and welded on the inside of the central headers. The remaining rows of tubes in this section are set in place row by row and welded to the headers.

The next section of header is then set in place as in figure 21b and welded to the first section and the flat plate. The tubes are then inserted and welded. These steps are repeated until the tube bundle is complete as in figure 22.

2. Exchanger Shell and External Headers

The ends of the exchanger shell are set in place and welded to the central headers. The four sections of the exchanger shell are set in place with the tube sleeves inserted in the proper holes in the exchanger shell. These sections are then welded together and to the end pieces. The tube sleeves are then welded to the shell as in figure 23.

The external headers are set in place and welded to the exchanger shell which then completes the exchanger erection and the exchanger is now ready to have the service piping attached. The final product is illustrated in figure 24.

To simplify illustration, the outer tube ends are shown as being in a line. However, with the tube sleeves nearly doubling the width of the slot in the shell into which the tube must fit, it is necessary to divide each row of tube ends into two lines as is shown in figure 23.

Central Header Construction

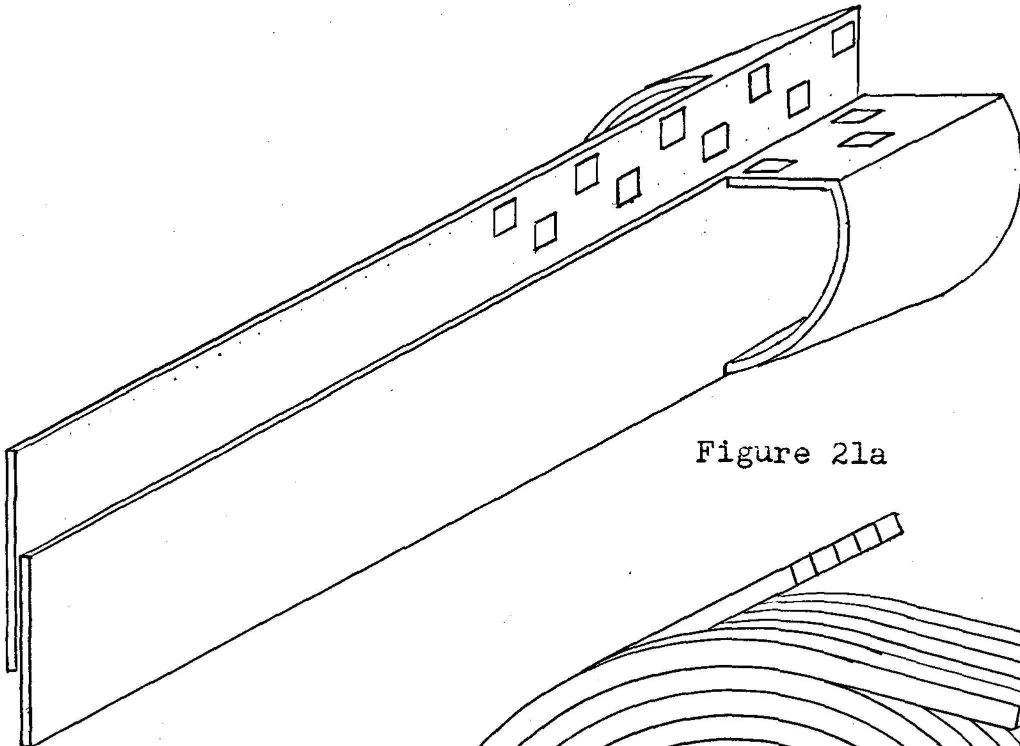


Figure 21a

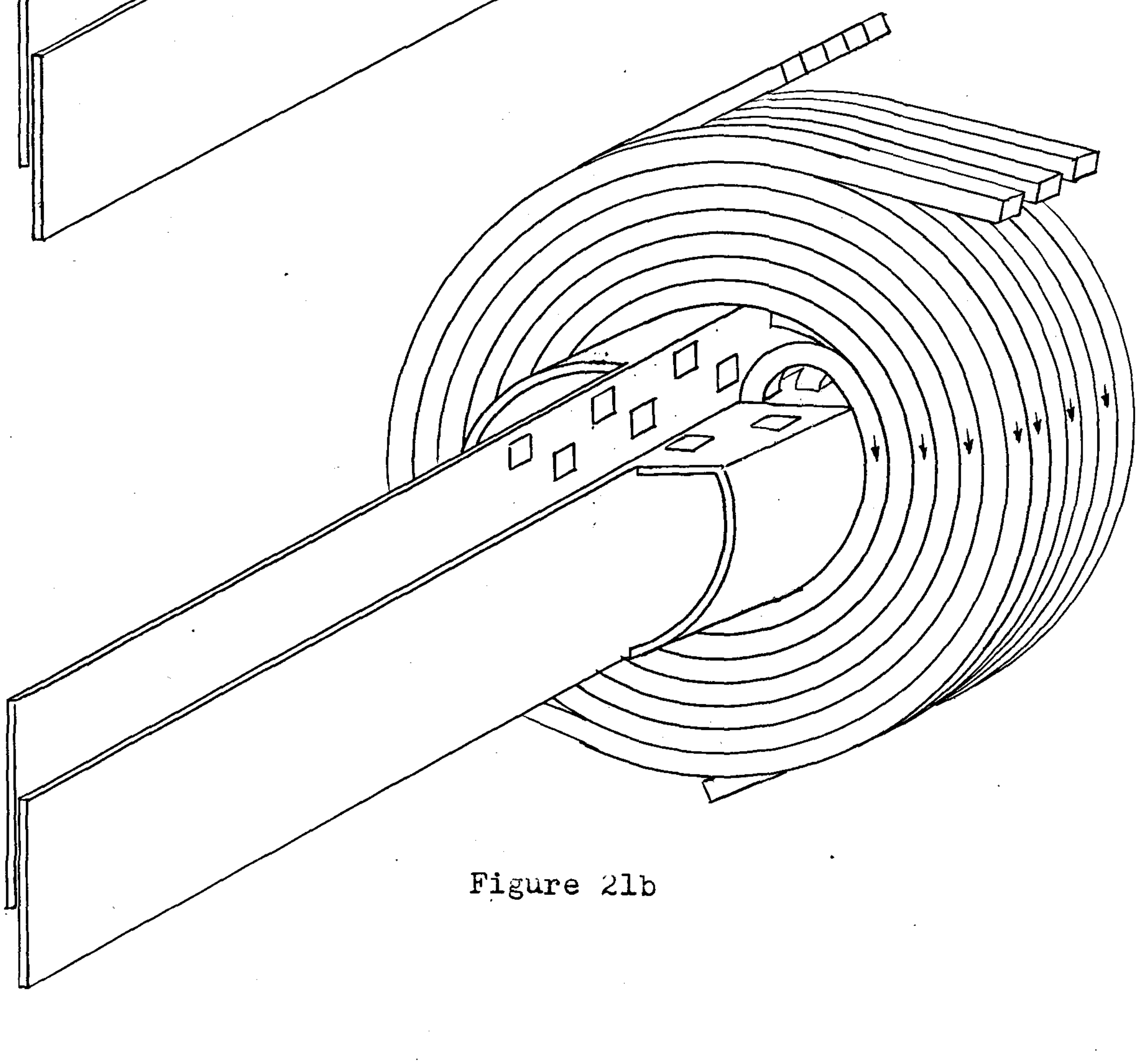


Figure 21b

Tube Bundle

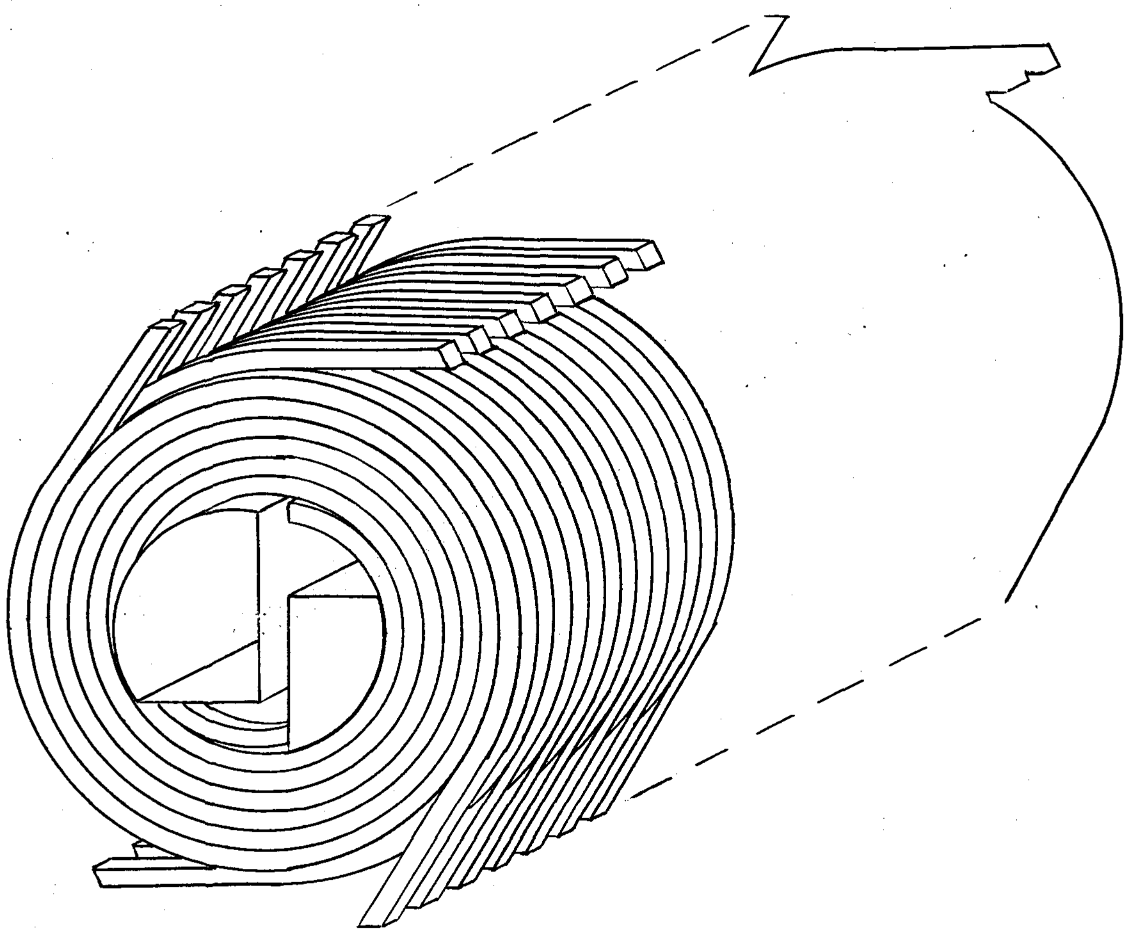


Figure 22

Exchanger Shell

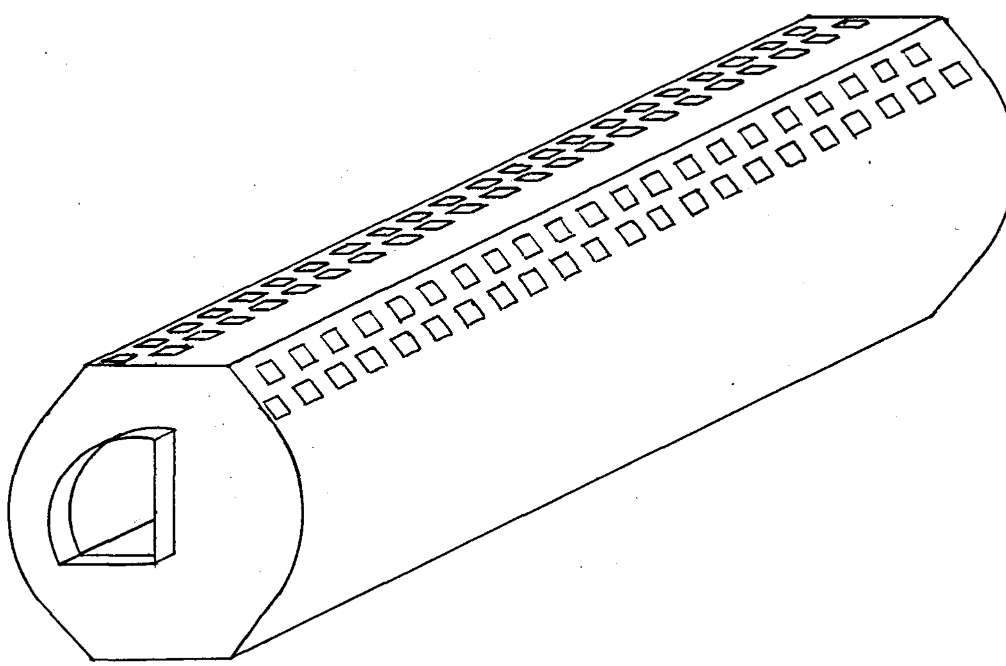


Figure 23

Completed Exchanger

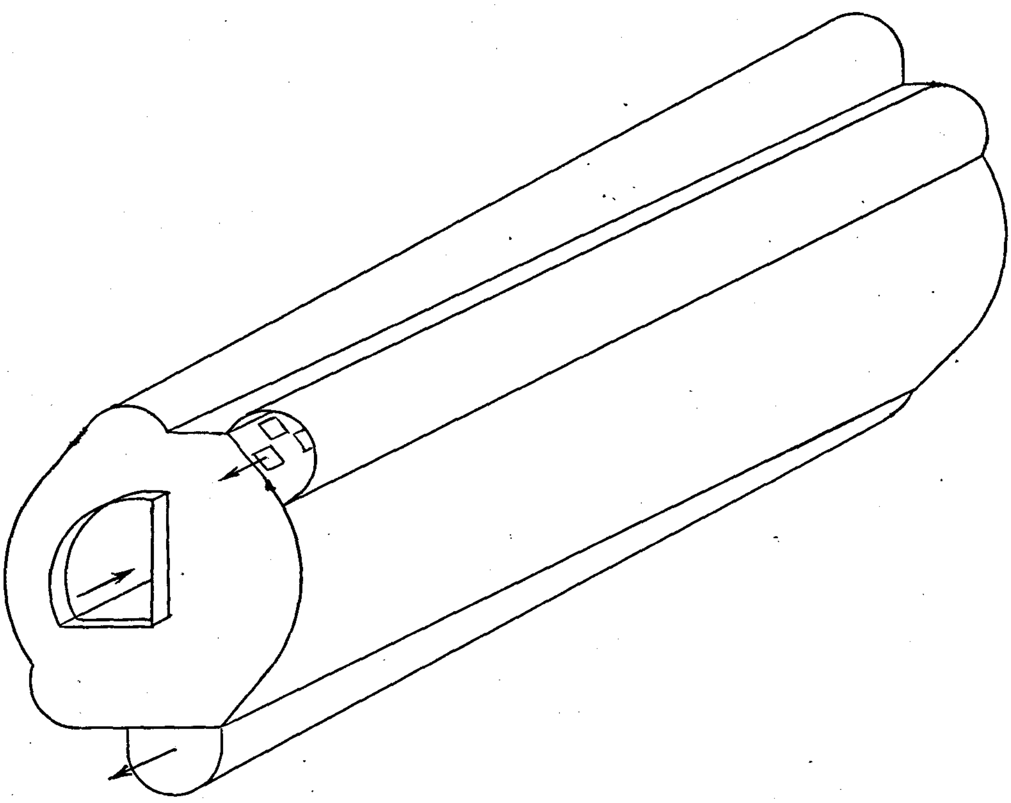


Figure 24

CHAPTER XI

CONCLUSIONS

1. Conclusions

The exchanger for which the method of construction has been outlined in all respects meets the specifications outlined in Chapter I. The overall volume of about 25 cu. ft. compares very favorably with other more conventionally designed exchanger volumes.

The construction of this exchanger entails more difficulties than most others. However, the slight increase in difficulty of construction is more than compensated for by the lack of thermal stresses which occur to a much greater extent in other exchangers and the slight reduction in volume.

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